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THE JOURNAL
OF
THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

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CONTENTS

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VOL. 32 JANUARY 1910 NUMBER 1

THE New York monthly meeting of the Society will be held in the Engineering Societies' Building on Tuesday evening, January 11. The subject for discussion is Lubrication. The paper upon Efficiency Tests of Lubricating Oils by Prof. F. H. Sibley of the University of Alabama, published in The Journal for November, will be presented and important contributions upon the properties of lubricants, their efficiency, durability, characteristics, etc., will be made by Dr. C. F. Mabery, of Case School, Cleveland, and Genl. Chas. Miller of Franklin, Pa.

Dr. Mabery has been engaged for a long period of time in experiments upon lubricating oils and has obtained results of unusual interest, because of the uniformity attained in repeating experiments, always a difficult matter in testing lubricants. General Miller has been so long identified with the subject of lubrication and has so large a fund of information as a result of this experience that his remarks will add greatly to the interest of the evening. There will be discussions also by F. R. Low, Editor of Power, I. E. Moultrop, mechanical engineer of the Boston Edison Company, J. P. Sparrow, chief engineer of the New York Edison Company, and others.

The subject of lubrication is so important in its bearing upon the conservation of power and upon machinery of all kinds, especially since the introduction of recent new types, such as the steam turbine and automobile, that it is desirable to have authentic information easily available for the use of engineers. By introducing the subject for discussion before the Society, it is hoped that this result may eventually be brought about and that a substantial beginning will be made at this meeting.

MEETING IN ST. LOUIS, JANUARY 15

The next monthly meeting of the Society in St. Louis will be held on January 15. The usual announcement of this meeting with details in regard to the paper and discussion will be sent to members and engineers in St. Louis and vicinity previous to the meeting.

MEETING IN BOSTON, JANUARY 21

A joint meeting of The American Society of Mechanical Engineers, the Boston Society of Civil Engineers and the Boston branch of the American Institute of Electrical Engineers, will be held in Boston on the evening of January 21. Committees have been appointed by the society of civil engineers and the local section of the electrical engineeıs to coöperate with the local committee of this Society to complete arrangements. The meeting will take the form of a banquet and reception, with the presidents of the three societies in attendance, George H. Westinghouse of The American Society of Mechanical Engineers, L. B. Stilwell of the American Institute of Electrical Engineers and George B. Francis of the Boston Society of Civil Engineers, besides the incoming president of the American Society of Civil Engineers, John A. Bensel, and other distinguished guests. The banquet hall of the Hotel Somerset, which is the largest and finest in the city has been engaged for the occasion.

Following the banquet there will be addresses by some of the guests and a paper on the Main and Auxiliary Machinery of the Battleship North Dakota, illustrated with lantern slides, by Charles B. Edwards of the Fore River Shipbuilding Company. There is under discussion at Boston a project for building and equipping a united engineering building and the president of the Boston Society of Civil Engineers will bring up this subject and describe what efforts that Society has already made towards this end.

The meetings of The American Society of Mechanical Engineers in Boston have been uniformly well attended, as have those of the other societies, and it is believed that this joint meeting will bring together an unusually large number of engineers and that it will be the most successful similar meeting of the kind that has taken place in that city.

SPRING MEETING, ATLANTIC CITY, MAY 31–JUNE 6

The Spring Meeting of The American Society of Mechanical Engineers will be held this year as usual, in addition to the London Meet-

ing which occurs in July. Atlantic City has been selected by the Meetings Committee and approved by the Council as the place and the time will be from May 31–June 6, inclusive. The headquarters during the meeting will be at the Marlborough-Blenheim Hotel.

JOINT MEETING OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS AND THE INSTITUTION OF MECHANICAL ENGINEERS

In response to the invitation of The Institution of Mechanical Engineers of Great Britain, received and accepted by The American Society of Mechanical Engineers, and recently sent out to the general membership, 133 members and 100 ladies have signified their intention of attending the joint meeting in Great Britain in the summer of 1910, and 183 have expressed themselves as giving the matter favorable consideration.

The present indications are that some of the functions will be held in Manchester, Birmingham or Sheffield, possibly concluding in London, and the invitation itself is an earnest of the notable professional and social opportunities which will be extended to the Society. Arrangements will probably be made for the accommodation of the members on the same steamer.

Where time and personal engagements permit, the visiting members will have the opportunity of attending the following events and meetings which are to take place during the summer of 1910: Anglo-Japanese Exhibition at Sheperds Bush, London; American Exposition in Berlin; Brussels Universal and International Exhibition; London Pageant, probably at Chester; Pageant at Bristol; Church Pageant at Fulham Palace; Military Pageant in London; International Congress of Mining, Metallurgy, Applied Mechanics and Practical Geology, at Düsseldorf; International Sports Exhibition at Vienna; International Exhibition of Arts and Industries, Alexandra Palace, London; the Passion Play at Oberammergau.

It is expected that papers will be presented by members of both societies on electrification of railways, on round house practice and the handling of locomotives at terminals, on certain phases of machine shop practice, and on the subject of standards for gear teeth which is now being considered by a committee of this Society as well as by a committee of the Institution of Mechanical Engineers. While papers will be mainly restricted to the subjects indicated, the Meetings Committee will be pleased to consider papers on other subjects.

REPORTS OF MONTHLY MEETINGS

BOSTON MEETING, NOVEMBER 17

A very successful meeting of the Society was held at Boston in the Lowell Building, Massachusetts Institute of Technology, Wednesday evening, November 17. Two hundred and forty were present at this meeting and the Low-pressure Steam Turbine was the topic of discussion.

Mr. Henry G. Stott of the Interborough Rapid Transit Company gave an interesting account of the difficulties encountered as well as the very fine results obtained from an installation recently made at the 59th Street Station of his company, New York. Mr. W. L. R. Emmet described the low-pressure turbine situation from his viewpoint and pointed out the advantages of this type of prime mover for many mill installations and industrial works in New England. Mr. H. E. Longwell, consulting engineer of the Westinghouse Machine Company, and Edward L. Clark, manager of their Boston office, both spoke on the work that company are doing in this field. Mr. Max Rotter, turbine engineer of the Allis-Chalmers Company, pointed out in a humorous way a number of situations where the low-pressure turbine was not a desirable proposition. Professor Miller of the Massachusetts Institute of Technology also discussed the subject.

BOSTON MEETING, DECEMBER 17

On Friday evening, December 17, a goodly number of engineers of Boston and vicinity gathered at the call of the local members of The American Society of Mechanical Engineers to discuss the Effect of Superheated Steam on Cast Iron. The meeting was called to order by Prof. Ira N. Hollis, who announced that the next meeting would be held on January 21 and would take the form of a reception, possibly a complimentary dinner, to the newly elected president of the American Society, George H. Westinghouse, and other of the Society's officials. A later announcement of this meeting is contained elsewhere in this number, and of the coöperation of the Boston Society

of Civil Engineers and of the local members of the Institute of Electrical Engineers. The committee which has been in charge of the meetings, consisting of Messrs. Hollis, Moultrop, Miller, Mann and Libbey, was continued.

The set papers which were published in the December issue of The Journal were then presented by their authors—Prof. Edward F. Miller of Boston, Arthur S. Mann of Schenectady, and Prof. Ira N. Hollis of Boston in the order named, and were discussed by Messrs. Collins of Stone & Webster, George A. Orrok of the New York Edison Company, Chas. H. Bigelow of Chas. T. Main's office, W. K. Mitchell of Philadelphia, Messrs. Primrose and Nutting of the Power Specialty Company, Wm. E. Snyder of the American Steel and Wire Company and others. The general purport of the discussion, was rather reassuring to the users of cast iron pipe and fittings, and to those who are interested in the extension of the use of superheated steam, in indicating that superheated steam *per se* has no injurious effect upon cast iron fittings, but that if the pipe lines are properly designed for the greater ranges of temperature, if the fittings are made adequate to the pressure and if fluctuations in temperature can be avoided, the use of superheated steam introduces no piping difficulties which can not be easily overcome.

MEETING AT ST. LOUIS, NOVEMBER 13

At the meeting of the Society at St. Louis, November 13, with the Engineers' Club at St. Louis, a description of the new plant of the Heine Safety Boiler Company of Boston was presented by E. R. Fish, under the title, A Modern Boiler Shop. There was also further discussion of Professor Carpenter's paper on High-Pressure Fire Service, continued from the October meeting.

MEETING AT ST. LOUIS, DECEMBER 11

A meeting was held with the Engineers' Club of St. Louis on Saturday evening, December 11, at the rooms of the latter society. The meeting was called to order by William H. Bryan, member of the Meetings Committee of the Society and chairman of the joint committee of the two societies at St. Louis. Prof. E. L. Ohle acted as secretary. There were present fifty-five members and guests.

The paper of the evening was by G. R. Parker of the General Electric Company, on The Relation of the Steam Turbine to Modern

Central Station Practice, in which the underlying principles of modern steam turbines were discussed, together with the design of various prominent types on the market, and the developments made in recent years in improving capacity and efficiency. Attention was called to the large turbine capacity which may now be obtained within limited floor space; to the question of low-pressure turbines and their availability in supplementing standard reciprocating engines, increasing both their capacity and economy; also to the work already done in this direction at the plant of the Union Electric Light & Power Company in St. Louis, and to prospective work along similar lines in the same plant. The address was illustrated by lantern slides.

Discussions followed by Chairman Bryan, Prof. H. W. Hibbard, L. R. Day, E. R. Smith and Prof. E. L. Ohle, in which many additional interesting points were brought out.

On the afternoon of the day of the meeting an excursion was made to the Ashley Street plant of the Union Electric Light & Power Company, for the inspection of the apparatus and equipment, on the invitation of John Hunter, chief engineer. This excursion, supplementing as it did the paper of the evening, added much to the interest and value of the meeting and a vote of thanks was extended to Mr. Hunter for the opportunity so generously afforded.

THE ANNUAL MEETING

The thirtieth annual meeting of The American Society of Mechanical Engineers was held in the Engineering Societies Building December 7 to 10, with an attendance of 628 members and 435 guests. This year, for the first time, the arrangements of the entertainment features were entirely in the hands of the local committee, the members in New York and vicinity acting as hosts, and the results fully justified this method of handling an important part of the annual meeting.

Despite the severe storm on Tuesday evening, the President's reception was well attended, and a large audience gathered in the auditorium. On Wednesday afternoon the trip through the Pennsylvania Terminal brought out a large body of members and guests, and in the evening L. W. Ellis, of the Bureau of Plant Industry, U. S. Dept. of Agriculture, delivered an interesting lecture on the Era of Farm Machinery. On Thursday evening the attendance at the reception in the magnificent ball room of the Hotel Astor was nearly 600.

OPENING SESSION, TUESDAY EVENING

The President's reception on Tuesday evening was undoubtedly one of the most enjoyable ever held, the members and guests comfortably filling the handsomely decorated rooms of the Society, though the inclement weather doubtless prevented a larger attendance.

The session was called to order in the auditorium by Vice-President Fred J. Miller, who presented President Jesse M. Smith. President Smith then proceeded with his address on The Profession of Engineering, which is printed in full in this number. It deals mainly with the need of coöperation among engineers, looking toward the maintenance of high standards in engineering practice.

Following the address, Theodore Stebbins, chairman of the Tellers of Election, presented to the President the report on the election of officers and the following were thereupon declared elected: For president, George Westinghouse; for vice-presidents, Charles Whiting

Baker, W. F. M. Goss, E. D. Meier; for managers, J. Sellers Bancroft, James Hartness, H. G. Reist; for treasurer, William H. Wiley.

President Smith then called on Past-Presidents Worcester R. Warner, Geo. W. Melville and Samuel T. Wellman to escort President-elect George Westinghouse to the platform.

After his notification of election and introduction to the members, the president-elect spoke as follows:

When Mr. Warner, the Chairman of your Nominating Committee, after first writing on the subject, came to Lenox to ask me to accept the nomination for president of this great Society, I had already decided that it would be impossible for me to have the privilege of accepting; but after he had explained to me the desires of his associates and had represented to me that it was the unanimous wish of all of the members of your Nominating Committee to honor me at this particular time, and in so doing to express an appreciation of my efforts and accomplishments in the engineering field, I with much hesitation consented to accept the nomination and promised if elected to do everything in my power.

Whether two mistakes have been made—one in yielding to the persuasive words of Mr. Warner, and the other in my election as your president—the forthcoming year will determine. I trust I may be able to fulfil your expectations by adding something to the worldwide reputation of The American Society of Mechanical Engineers.

With these remarks, I now accept with feelings of deep gratitude the honor which the members of the Society have tonight unanimously conferred upon me.

There never was a time in the history of the world when honest, wise and conservative action is more strongly demanded of us and of all men than now, if we have any desire to preserve the right to comfortably carry on our various affairs.

I thank you, and I ask your coöperation in my efforts to perform my duties as your president.

The meeting was then adjourned to the rooms of the Society where the members and guests were introduced by Secretary Calvin W. Rice, to the President-elect and Mrs. Westinghouse, those also in the receiving line being President Jesse M. Smith and Mrs. Smith, Mrs. Hutton and Honorary Secretary F. R. Hutton.

WEDNESDAY EVENING LECTURE

As already stated the lecture on Wednesday evening was on the Era of Farm Machinery, by L. W. Ellis, of the Bureau of Plant Industry of the United States Department of Agriculture at Washington, D. C. The lecture was illustrated by lantern slides. Mr. Ellis first gave an idea of agricultural progress, by describing some of the most striking mechanical achievements found on Western farms of the present day. He first described early farm implements and told briefly of the

transition from hand to machine methods. In 1800 wheat was sown broadcast by hand, after the ground had been plowed with a heavy, clumsy, wooden plow, requiring as many as eight oxen to pull it. Sickles cut the grain, and it was bound by hand. During the succeeding winter it was threshed out either by a flail or by driving animals over it as it lay in heaps. It was finally winnowed by hand.

Corn cultivation was by the hoe, or a rude shovel plow. The stalks were cut and the ears husked out by hand. Shelling was done by scraping the ears against the handle of a frying pan—a bushel in one hundred minutes.

Hay was cut with a scythe and was pitched by hand from ground to cart, and cart to haymow. Baling and shipping were practically unknown. Hand methods prevailed in the dairy, the stable, the cotton fields, the potato patch—in fact in every phase of production.

From 1855 to 1894 the human labor consumed in producing a bushel of corn by the best available methods declined from four hours and thirty minutes to forty-one minutes, and for shelling it from one hundred minutes to one minute. In 1830, three hours and three minutes of human labor were required to raise and thresh a bushel of wheat—in 1896 ten minutes. Eleven hours were required to cut and cure a ton of hay in 1860, and but one hour and thirty-nine minutes in 1894.

Power corn shellers now used have a capacity of from one hundred to eight hundred bushels per day. The cobs are carried to a pile and the shelled corn delivered into sacks or wagons. The fuel value of the cobs pays the cost of shelling.

Though hand methods still prevail in some sections, the mower is now practically the universal means of cutting the hay crop. This is a modification of the early reaping machines with such factors eliminated as are not necessary for cutting the grass. The steel self-dump rake, the side-delivery rake and the hay loader, the stacker, and the bailing press are other developments for hay harvesting.

In the extreme West there has been developed the combined harvester which seems to represent the greatest possible saving of human labor. This machine, drawn by from twenty to forty horses, under control of a single driver, cuts, threshes, recleans, and delivers into sacks the grain from forty to fifty acres per day. Two men are required for sewing the sacks. The straw, including all weed seeds, is distributed over the ground as the team proceeds. On level land the horses may be replaced by the steam engine, which furnishes power sufficient to cut a swath up to forty feet in width and to cover from seventy-five to one hundred and twenty-five acres per day.

For general farm work the internal-combustion tractor may be said to be rapidly supplanting the steam engine, which, however, has a great field of usefulness in sections where it is desired to bring large areas rapidly under cultivation. In older sections, in order to compete successfully with the horse, tractors must bring the cost of operation close to the cost with horses and at the same time be capable of a great variety of work. The internal-combustion tractor meets these conditions better than the steam engine, and is being introduced at a rate estimated anywhere from two thousand to five thousand per year.

The automobile is rapidly finding a place in the business management of the farm. It takes from the heavy draft horse the necessity for long, exhausting trips to town on light errands.

In general, machinery has reduced the cost of producing farm products. It has improved the quality of products by condensing crop operations within the period when the most favorable conditions prevail. By increasing the acre effectiveness of a man it has reduced the labor necessary to produce the nation's food supply, leaving it free to assist in development along other lines. At the same time it has thrown upon the cities the burden of providing work for an ever increasing army of non-producers. It has increased the investment necessary for the proper organization of a farm, this and the price of land making it more difficult for a person of small capital to engage in farming.

As a nation we have occupied nearly all of our naturally productive area and are confronted with the necessity of providing food for an increasing population with a constant acreage. In the past, machinery has encouraged extensive rather than intensive farming. Henceforth the reverse should be true. If he who makes two blades of grass grow where one grew before, is a public benefactor, then none the less is he a public servant who puts into the farmer's hands the machinery for making such a course attractive.

BUSINESS MEETING

The business session on Wednesday morning was called to order by President Jesse M. Smith. Secretary Calvin W. Rice read the annual report of the Council. The Secretary then read the report of the Tellers of Election of members, which will be published in the membership list of the Society. The list included 166 applicants for membership and 21 for advance in grade.

The next in order was the consideration of the proposed amendments to the Constitution. The first amendment relates to C 10 on associate membership, which reads as follows:

C 10 An Associate shall be 26 years of age or over. He must either have the other qualifications of a member or be so connected with engineering as to be competent to take charge of engineering work, or to coöperate with engineers.

The proposed amendment reads as follows:

An associate member shall be thirty years of age or over; he must have been so connected with some branch of engineering, or science, or the arts, or industries, that the Council will consider him qualified to coöperate with engineers in the advancement of professional knowledge.

Another amendment relates to the clause on Junior Membership which now reads as follows:

C 11 A Junior shall be 21 years of age or over. He must have had such engineering experience as will enable him to fill a responsible subordinate position in engineering work, or he must be a graduate of an engineering school.

The following addition is proposed by the Committee on Constitution and By-Laws:

A person who is over 30 years of age can not enter the Society as a Junior.

Both these amendments have been approved by the Committee on Membership. It therefore remains for the members to vote on them by letter ballot.

A third proposed amendment to the Constitution relates to the formation of an additional standing committee. This was presented at the Washington meeting in the form of a resolution, as follows:

Resolved, That we recommend to the Council the appointment of a Public Relations Committee, to investigate, consider and report on the methods whereby the Society may more directly coöperate with the public on engineering matters and on the general policy which should control such coöperation.

It was moved and seconded that this also be referred to the members for letter ballot.

Dr. D. S. Jacobus, Chairman of the Committee on Power Tests, then made a verbal report. This committee was appointed to revise all the codes relating to power tests, some of which did not agree with others, or were not up to date. It had been decided to blend the whole into one report rather than present a series of reports, as on engine testing, boiler testing, etc. The first part of the report will deal with tests in general, calibration of apparatus, units, etc., while

the second part will be subdivided for the various classes of machines and apparatus.

Geo. H. Barrus had volunteered to prepare a skeleton of the report and had done excellent work in this respect, the material making 69 closely type-written pages. Copies of this outline were in the hands of the members of the committee and would shortly be discussed by them.

Dr. Jacobus also made a verbal report for the Joint Committee on a Standard Tonnage Basis for Refrigeration. This committee had made a preliminary report in 1904 and suggested certain units for measuring the refrigerating capacity of the machinery. They had also suggested a standard set of conditions under which a machine should be tested to obtain the refrigerating capacity of that machine. Later on, the work of the committee was extended, and they were asked to recommend a method of testing the machines. A preliminary report was also prepared on this portion of the work and had been before the Society.

Though the committee had received some favorable discussion on the report they felt that it was not a complete piece of work, and they wished that some one would give the committee additional light on how the report could be made. Furthermore, there were many places in the report where the committee could not make any definite recommendations, because they did not have enough data at hand.

A résumé of the work that has been done by the Committee on Refrigeration was prepared and sent to the Congress of Refrigerating Industries, held in Paris in the fall of 1908, with the request that it be discussed. In making this résumé certain questions were asked, on which the committee wished to obtain specific information. This was done in a semi-official way, and after taking up the matter with the Secretary of this Society, the committee ended the communication to the International Committee in this way:

The policy of The American Society of Mechanical Engineers has always been for the advancement of the arts, and whereas it is only natural that it should take pride in participating in advancements, it will never look except with satisfaction upon activities of other bodies, even in the subjects on which it has worked.

I feel safe in saying, therefore, that any criticism by the members of this organization on the work which has been done in connection with the subject at hand will be gladly received. Criticism leads to the establishment of better and more up-to-date methods, and what The American Society of Mechanical Engineers is after, and what I am sure we are all after, is to work hand in hand for the good of the cause.

I also feel safe in saying that The American Society of Mechanical Engineers

will coöperate in every way in the endeavor to establish some standard set of rules which shall conform with the views of such able experts as are gathered in this meeting. It is certainly hoped that the matter presented in this paper will receive a thorough discussion, irrespective of whether those who take part agree or disagree with the findings of the committee.

About the same time, a request was made by the committee that it should be allowed to coöperate with a committee of the American Society of Refrigerating Engineers, so that if this general committee recommended certain units, they would really be used by both Societies. A committee of five was appointed by the American Society of Refrigerating Engineers to coöperate with the committee of five of The American Society of Mechanical Engineers. This combined committee has already held one meeting and sent out a circular letter to a number of refrigerating engineers, reviewing the units that had been recommended by the Society, and asking for an opinion regarding these specific units. A great number of replies had been received, showing how much interest there is in the subject. Most of the replies said either that the units were acceptable to those who had read the letter, or that they would leave the selection of the units entirely in the hands of the committee. The committee therefore has a very good working basis, and hopes within a comparatively short time to be able to present the results of its work.

Dr. C. E. Lucke then abstracted the report of the Gas Power Standardization Committee, of which he is chairman. The report was discussed by Dr. D. S. Jacobus, Prof. R. H. Fernald, A. A. Cary, Edwin D. Dreyfus and L. B. Lent.

The report of the Gas Power Plant Operations Committee was presented by F. R. Low in the absence of I. E. Moultrop, chairman of the committee. The report was discussed by Prof. R. H. Fernald, Edwin D. Dreyfus, and Arthur J. Wood.

THURSDAY MORNING SESSION

The Thursday morning session was devoted to papers on the measurement of the flow of fluids.

The first paper presented was on Tests on a Venturi Meter for Boiler Feed, by Prof. C. M. Allen, of Worcester Polytechnic Institute. The object of these tests with the venturi meter was to determine how well adapted it would be for use in measuring the feed to a boiler, in view of the variety of conditions under which it might have to operate. The methods of pumping the water through the meter, the different temperatures of the water pumped, various and fluctuating

pressures and velocities of flow, any one or several of these conditions might be met in actual service, and the results obtained indicate that such occurrence would have practically no effect on the satisfactory performance of the work of the meter. Though there are limits to the satisfactory operation of any one meter, the tests indicate that the venturi meter is sufficiently accurate for the majority of commercial or engineering requirements.

The paper was discussed by F. N. Connet and Clemens Herschel, Dr. Sanford A. Moss and Prof. L. S. Marks submitting written discussions.

The next paper, Efficiency Tests of Steam Nozzles, by Prof. F. H. Sibley, of the University of Alabama, was read by Prof. C. C. Thomas of the University of Wisconsin. The object of the test was to determine the efficiency of various shaped nozzles with steam flowing from a given initial pressure to a known vacuum; also to determine the effect on the efficiency of changing the angle of divergence. Two methods were tried out for finding this efficiency: (*a*) by first finding the pressure in the nozzle by means of a search tube placed axially in the nozzle; (*b*) by finding the reaction of the nozzle by suspending it in an air-tight box at the end of a flexible steel tube. The deflection of the tube caused by the reaction of the nozzle was measured by a calibrated spring. The results of the tests indicate: (*a*) that the reaction is affected by a difference in pressure between the muzzle of the nozzle and the medium surrounding the nozzle; (*b*) that the efficiencies of the various nozzles were determined within a probable error of 2 per cent; (*c*) that the efficiency is affected more by the smoothness of finish on the inside of the nozzle than by the exact contour of the nozzle.

A. F. Nagle, A. R. Dodge and Professor Thomas discussed the paper, J. A. Moyer submitting a written discussion.

George F. Gebhardt's paper on The Pitot Tube as a Steam Meter was read by the Secretary in the author's absence. The application of a pitot tube system as described in the paper is an accurate means of determining the *velocity* of steam at any point in a pipe, provided the values of the various influencing factors are known; and for straight lengths of piping with continuous flow, under these conditions, it is an accurate means of determining the *weight* of steam flowing. Under average commercial conditions in which the pressure and quality of the steam fluctuate and an average value must be taken for the density of the self-adjusting water column, only approximate results can be obtained, the extent varying with the degree of fluctuation.

Walter Ferris and A. R. Dodge discussed the paper, a written discussion by Prof. W. B. Gregory being read by the Secretary.

The paper on An Electric Gas Meter was presented by the author, Prof. Carl C. Thomas, of the University of Wisconsin. The paper describes a meter measuring the rate of flow of gas or air, which can be adapted for use as a steam meter or as a steam calorimeter. The operation of the gas meter depends upon the principle of adding electrically a known quantity of heat to the gas and determining the rate of flow by the rise in temperature of the gas (about 5 deg. fahr.) between inlet and outlet. The adoption of this principle of operation permits the construction of a very accurate and sensitive autographic meter of large capacity containing no moving parts in the gas passage; independent of fluctuations in pressure and temperature of the gas; and capable of measuring gas or air at either high or low pressures or temperatures. The electrical energy required is about 1 kw. per 50,000 cu. ft. hourly capacity, at the pressures ordinarily used in gas mains.

Prof. W. D. Ennis, E. D. Dreyfus and A. R. Dodge discussed the paper, a written discussion from Prof. L. S. Marks being read also.

THURSDAY AFTERNOON—STEAM ENGINEERING

At the Thursday afternoon session Vice-President L. P. Breckenridge presided. Five papers were presented dealing with different phases of steam engineering. The first paper, Tan Bark as a Boiler Fuel, by David M. Myers, described the results obtained by burning spent hemlock tan bark, the average fuel value of which is about 9500 B.t.u. per lb. of dry matter, which is about 35 per cent of its total moist weight in the fireroom. The available heat value per pound as fired is 2665 B.t.u. One ton of air-dry hemlock bark produces boiler fuel equal to 0.42 tons of 13,500 B.t.u. coal. A. A. Cary, Prof. Wm. Kent and Prof. L. P. Breckenridge took part in the discussion.

J. R. Bibbins then presented his paper on Cooling Towers for Steam and Gas-Power Plants, which contained a critical study of different types of towers with a description of their distinctive features. The paper also describes a simple inexpensive type of tower employing a lath-mat cooling surface and offers suggestions for a combination of natural-draft and forced-draft types.

The paper was discussed by Geo. J. Foran, W. D. Ennis, H. E. Longwell, B. H. Coffey, E. D. Dreyfus and F. J. Bryant. A written discussion by Carl G. de Laval was read by the Secretary.

W. P. Caine's paper, Governing Rolling Mill Engines, was read by Richard H. Rice. The paper describes and gives indicator cards and speed curves of a Corliss engine driving a three-high mill under two different conditions of governing, (*a*) under the widest range of adjustment of cut-off, (*b*) under a limited range, increasing the economy and making the engine run much more smoothly and safely. A table gives the power required for rolling in the mill and the momentary source of energy, whether from the cylinder or flywheel. A description is also given of the tachometer used to take the speed curves. Written discussions by H. C. Ord and James Tribe were read by the Secretary.

The next paper was that by F. W. Dean on An Experience with Leaky Vertical Fire-Tube Boilers. The author discussed the difficulties experienced with some large vertical boilers, somewhat over 10 ft. in diameter, and containing over 6000 sq. ft. of heating surface. The boilers leaked badly very soon after being started and nothing that was done improved their condition until the water legs were lengthened from 2 ft. to 7 ft. 2¾ in., the boilers thus being raised 5 ft. 2¾ in. Before they were raised the lower ends of the tubes would cover with very hard clinker and become stopped up. This clinker could be removed only by cutting it off when the boilers were cold. After the boilers were raised, a light clinker that could be blown off formed about the tubes; by removing this by blowing every three or four hours the leaks were stopped and they have never returned.

Those taking part in the discussion were R. P. Bolton, Prof. Wm. Kent, J. C. Parker, O. C. Woolson, A. A. Cary, Prof. A. M. Greene, Jr., E. D. Meier and D. M. Myers. A. Bement submitted a written discussion.

Mr. Dean's second paper, The Best Form of Longitudinal Joint for Boilers, dealt with the defects of the usual form of butt joint used on the longitudinal seams of boilers, in which the inside strap is wider than the outside strap. It gave some history of the joint and discussed some of its defects and suggested a substitution for this form.

The paper was discussed by R. P. Bolton, Carl G. Barth, E. D. Meier, Prof. A. M. Greene, Jr., W. A. Jones, Prof. S. W. Robinson, Geo. I. Rockwood, and Sherwood F. Jeter.

GAS POWER SECTION

The session of the Gas Power Section was held on Thursday afternoon, Chairman F. R. Low presiding. In his address, the Chairman

referred briefly to the work of the various committees of the Section and stated that during the year the membership had increased from 247 to 378, a gain of over 50 per cent. Mr. Low also dealt with the development in the gas-power field during the year, mentioning some experiments with gas turbines. Gas-engine design, the use of by-product gases, the development of the bituminous producer, the gasification of peat, and the gas engine in marine work, were also briefly dealt with.

The report of the Tellers of Election, Edw. Van Winkle, Prof. Walter Rautenstrauch and J. V. V. Colwell, was then presented by Prof. Rautenstrauch, the results being as follows: for chairman J. R. Bibbins 107; for member of the Executive Committee, F. R. Low 108.

The report of the Gas Power Plant Operations Committee was then presented by James D. Andrew, and discussed by J. C. Parker, J. N. Norris and H. H. Suplee. Prof. C. H. Benjamin reported verbally for the Literature Committee, outlining the work of the committee in bringing gas-power literature to the attention of the members. H. R. Cobleigh and Professor Rautenstrauch also spoke on the work of this committee, the latter suggesting a plan for better organization of the committee to deal with literature on the subject.

L. B. Lent reported for the Gas Power Installations Committee that two forms had been prepared and sent to manufacturers, and while a good deal of information had been received, not enough was on hand for a complete report. The committee hoped to have the material in shape at an early date.

Prof. W. F. M. Goss then presented the paper on Testing Suction Gas Producers with a Koerting Ejector, by C. M. Garland and A. P. Kratz. The paper describes a method of testing the suction gas producer which is independent of the engine. The engine is blanked off from the producer and a Schutte & Koerting steam ejector is inserted, which draws the gases from the producer and delivers them to a scrubber in which the steam used by the ejector is condensed. The gases then pass to a meter for measuring their volume. Complete data of calculations and results are given in appendices.

The paper was discussed by Prof. R. H. Fernald, G. M. Tait, H. H. Suplee, L. B. Lent, S. C. Smith, W. B. Chapman and Edw. N. Trump.

The paper on Bituminous Gas Producers was then presented by the author, J. R. Bibbins. The paper describes a double-zone type of producer and the results obtained in gasifying bituminous coal. Continuous operation was secured with tar-free gas of reasonable heat value and producer efficiency and an over-all plant economy of about

one pound of fair bituminous coal per brake horsepower (proportionate economies for poorer grades). The efficiency and general effectiveness of operation of the producer on low-grade fuel, lignites, etc., was practically as high as with the higher grades. The following took part in the discussion: G. M. Tait, Prof. R. H. Fernald, W. B. Chapman, H. M. Latham, H. H. Suplee, Edw. N. Trump, H. B. Langer, S. C. Smith, Prof. W. Rautenstrauch, and G. D. Conlee.

FRIDAY MORNING

The session on Friday morning opened with the paper by Walter Ferris on The Bucyrus Locomotive Pile Driver. This paper describes a new railway pile driver, the leading feature of which is a very powerful propelling apparatus and a large boiler, enabling it to act as a locomotive and haul its own train of tool cars, boarding cars, etc., over the road. A special turn-table, consisting of hydraulic lifting apparatus and a large ball-bearing, enables the entire pile driver, including trucks, to be turned end for end or crosswise of the tracks. O. K. Harlan discussed the paper, A. F. Robinson and L. J. Hotchkiss submitting written discussions.

The paper by Henry Hess on Lineshaft Efficiency, Mechanical and Economic, described the test of the relative efficiency of a lineshaft of $2\frac{7}{16}$ in. diameter, making 214 r.p.m., with bearing load due to the weight of the parts plus the tension of the belts subjected to known stress by counterweighting, when running in ring-oiling babbitted bearings and when mounted in ball bearings. The savings in power consequent on this change ranged from 14 to 65 per cent, with 36 and 35 per cent under average conditions of good practice, due to belt tensions of 44 lb. and 57 lb. per inch width of single belt respectively. The paper gives data for determining the power savings that may be expected in various plants, by the use of ball bearings.

Those discussing the paper were T. F. Salter, Prof. R. C. Carpenter, C. A. Graves, O. K. Harlan, C. J. H. Woodbury, Walter Ferris, Fred J. Miller, A. C. Jackson, C. D. Parker and Oliver B. Zimmerman. Geo. N. Van Derhoff submitted a written discussion.

A. F. Nagle's paper on Pump Valves and Valve Areas, called the attention of engineers to the need of reviewing the common notion that "valve-seat area" is synonymous with "velocity of flow." The purpose of specifications for pumping engines is to secure a low velocity of flow through the valves, thus reducing the head required to force water through the pump; but to accomplish this purpose, special

and intelligent attention should be given to the springs of the valves, rather than to valve-seat areas. If that be done, valve-seat areas need not be greater than the plunger area for the vertical triple-expansion pumping engines so largely used in city pumps. Prof. W. M. Kent, A. B. Carhart, Prof. R. C. Carpenter and E. H. Foster discussed the paper. Contributed discussions were by Chas. A. Hague, I. H. Reynolds and F. W. Salmon.

Another paper by Mr. Nagle, a Report on Cast-Iron Test Bars, brought out the fact that test pieces, whether cast in separate molds or in the same mold as the main casting, are not perfect indications of the character of the iron in the main casting. The results obtained by the author would indicate a probable variation of 15 per cent where uniformity might be expected. A. A. Cary and T. M. Phetteplace discussed the paper, contributed discussion being by Prof. W. B. Gregory and Geo. M. Peek.

The meeting closed with the following resolutions, offered by Luther D. Burlingame:

Whereas The American Society of Mechanical Engineers at its Annual Meeting, December 1909, desires to express its appreciation to those who have provided opportunities for entertainment and on behalf of the visiting members and their guests thanks for the cordial welcome extended by the local members and their friends of New York and vicinity,

Be it ***Resolved*** that the Secretary extend the thanks of the Society and express the appreciation of its members and guests to the local committee for their untiring efforts, to those who have sent invitations to visit technical and engineering works and places of interest, to Mr. Geo. Gibbs, chief engineer of the Pennsylvania Tunnel and Terminal Railroad Co., and to Mr. Walter Kerr, president of the Westinghouse, Church, Kerr & Co., and their associates, for the opportunity to inspect the new Pennsylvania Railroad station; to Dr. B. T. Galloway, chief of the Bureau of Plant Industry, Department of Agriculture, for the very instructive and entertaining paper on The Era of Agricultural Machinery, and especially to those ladies who have so efficiently assisted by extending a generous hospitality to their guests.

EXCURSIONS

As usual at Conventions of the Society there were numerous excursions to points of interest in New York and vicinity, which constituted an important feature of the program for the entertainment

of visiting members and guests. Invitations for these excursions were generously extended by many firms and individuals, and through the efforts of the Excursion Committee, Hosea Webster, *Chairman*, trips to various plants and industries were arranged, to the representatives of which the grateful appreciation of the Society has been expressed.

A list of excursions follows:

Pennsylvania Railroad Terminal and Passenger Station: Invitation by George Gibbs, Chief Engineer, Pennsylvania Tunnel Terminal R. R. Co., and member of the Society; Henry R. Worthington Hydraulic Works, Harrison, N. J., by William Schwanhausser, Chief Consulting Engineer of International Steam Pump Co., member of the Society; Harrison Lamp Works of General Electric Co., Harrison, N. J., by George H. Morrison, General Manager; Interborough Rapid Transit Co., central power station at 59th St., New York, by H. G. Stott, Superintendent of Motive Power, Manager of the Society; Edison factories and Edison Laboratory at Orange, N. J., by Frank L. Dyer, President of National Phonograph Co., associate member of the Society; De La Vergne Machine Co., New York, by Adolf Bender, President; New York Telephone Co.; Gramercy and Stuyvesant Central Offices, by E. F. Sherwood, Superintendent of Traffic; Crocker-Wheeler Co., Ampere, N. J., by S. S. Wheeler, President, member of the Society; Westinghouse Lamp Co., Bloomfield, N. J., by Walter Carey, General Manager; New York Edison Co., Waterside Stations Nos. 1 and 2, by John W. Lieb, Jr., 3d Vice-President, member of the Society; Astoria Light, Heat & Power Co., Astoria, N. Y., by William H. Bradley, Chief Engineer, Consolidated Gas Co., member of the Society; Brooklyn Rapid Transit Co., Williamsburg Power Station, by C. E. Roehl, Electrical Engineer; Rockland Electric Co., Hillburn, N. Y.; Singer Building, New York, by Singer Mfg. Co.; Trenton Iron Co., Trenton, N. J.; Watson-Stillman Co., Ampere, N. J.; Metropolitan Life Insurance Building, New York.

Every possible courtesy was extended to the visiting parties in each case and in some instances special transportation facilities were provided. At the Edison Laboratory visitors were met by Thomas A. Edison, Hon.Mem.Am.Soc.M.E., who personally explained many points of interest about the plant. In order to avoid confusion, arrangements were made to assemble at the various manufactories at a time and place indicated in the program. The Information Bureau, located in the foyer of the building, under the chairmanship of F. E. Idell, was of material aid in this connection.

ENTERTAINMENT FEATURES

The Ladies' Reception Committee, composed of ladies resident in and about New York, under the chairmanship of Mrs. Herbert Gray Torrey, contributed much to the pleasure of members and guests of the Society. Tea was served from four until six o'clock on Tues-

day, Wednesday and Thursday afternoons during the convention, in the ladies' headquarters in the reception rooms of the Society on the eleventh floor. Mrs. George H. Westinghouse was the guest of the committee on Wednesday afternoon.

A number of excursions to shops and hotels were arranged and successfully carried out under the guidance of members of the committee. The kindness of Mr. and Mrs. John W. Lieb, Jr., made possible several enjoyable automobile rides through Central Park and Riverside Drive.

MEETINGS OF THE COUNCIL

DECEMBER 7, 1909

A meeting of the Council was called to order December 7, 1909, in the rooms of the Society, with President Smith in the chair. There were present at the meeting Geo. M. Basford, Geo. M. Bond, L. P. Breckenridge, R. C. Carpenter, H. L. Gantt, A. C. Humphreys, F. J. Miller, A. M. Waitt, Past-Presidents Charles Wallace Hunt, F. R. Hutton, Ambrose Swasey, F. W. Taylor and S. T. Wellman, and Calvin W. Rice, Secretary. The Council was especially pleased to have present John Fritz, Honorary Member and Past-President.

The minutes of the previous meeting were read and approved. The Secretary reported the deaths of Charles H. Willcox and William Metcalf.

The amendments to By-Laws B-6, B-7, B-12, B-13, B-18, B-19, B-27, B-28, B-34, and B-36 and the new By-Laws, one providing for the appointment of a Trustee of the United Engineering Society and one respecting The Journal of the Society, were approved.

EXECUTIVE COMMITTEE

Voted: That the Council sees no objection to any group of members selecting their own fiscal agent or correspondent, through whom the transmittal of their dues and other indebtedness to the Society may be made.

Voted: To refer the communication of the Western Society of Engineers, regarding the revision of the building laws of the State of Illinois, to the Public Relations Committee, to be appointed.

Voted: To approve the exchange of house and library privileges with the Louisiana Engineering Society.

The Secretary reported that circulars regarding the Joint Meeting in England had been issued to the membership and 116 favorable replies had already been received.

Voted: That the Council approve the recommendation of the Executive Committee approving coöperation with the Association of

American Steel Manufacturers to secure the general adoption of a system approved by a committee of the Society, December, 1894 (Trans., vol. 16, p. 32), to call the thickness of metals by their dimensions in decimals of an inch rather than by arbitrary number, and the Council recommends that the President appoint a committee to cöoperate with the Association.

The following were constituted such a committee: S. T. Wellman and George M. Bond.

The resolution referred to the Council from the Washington meeting, regarding the increase of facilities of the United States Patent Office, was laid on the table.

FINANCE COMMITTEE

The following resolutions were received from the Finance Committee and on motion approved:

That the Finance Committee recommend to the Council that the transfer of 10 per cent of the Reserve Fund of the Current Income Account be discontinued, as recommended in the Annual Report.

That the Secretary be authorized to charge against this Annual Meeting Sub scription Fund, namely, $205.60, whatever bills may have been incurred by the office in behalf of the Local Committee for the Annual Meeting, and the balance, if any, be paid by the Treasurer to the Local Committee of 1909.

Voted: To approve the recommendation of the Finance Committee that a committee be appointed to take up the consideration of the question of increasing the membership and providing ways and means to put the same into effect during the coming year.

LIBRARY COMMITTEE

Voted: To adopt the following resolutions of the Library Committee but with the amendment that the House Committee have the first option on duplicate books, to enable that Committee to furnish the reception room:

To recommend to the Council that the Librarian be authorized to sell to the highest bidder, for the benefit of the Society, the duplicate books recommended in the letter of the Librarian to the Committee, dated July 23, 1909.

MEETINGS COMMITTEE

Voted: To receive the resolution of the Meetings Committee, to whom had been referred the action of the Council on a Machine Shop Section, but to amend to read:

Voted: To advise the Council that the Committee is in accord with the plans of the Council for carrying out the purpose of a Machine Shop Section through committees appointed by the Meetings Committee and not by the formation of a special section; and that the Committee will proceed to such plans as soon as possible.

Voted: To approve the recommendation of Atlantic City for the Spring Meeting of the Society, May 31 to June 3, 1910.

STUDENT BRANCHES

Voted: On recommendation of Professor Hutton, Chairman of the Sub-Committee on Student Branches, to approve the applications of the University of Nebraska at Lincoln, Neb., and the University of Missouri at Columbus, Mo., to form student branches of the Society.

A communication was read by the Secretary regarding the possibility of holding meetings of the Society in Chicago, along the lines of those in St. Louis and Boston.

The Secretary presented a draft of the annual report of the Council which after amendment, was approved and ordered filed and printed as the report of the Council for 1909.

Voted: That the Library Committee be requested to give consideration to the question of procuring and caring for a collection of lantern slides.

THURSTON MEMORIAL

Dr. Humphreys reported the intention of the Thurston Memorial Committee to have the dedication exercises in February at the regular monthly meeting of the Society, and requested suggestions from the Council of suitable speakers for that evening, covering the various phases of Dr. Thurston's life work at the Naval Academy, Stevens Institute and Cornell University, as well as his laboratory and research work and work in connection with the organization of the Society.

The meeting adjourned.

DECEMBER 10, 1909

A meeting of the Council was called to order by Jesse M. Smith, Past-President, on December 10, 1909, in the rooms of the Society.

Mr. Smith appointed Vice-President R. C. Carpenter and Manager I. E. Moultrop a committee to introduce to the Council the Vice-Presidents-elect and Managers-elect, and Past-Presidents Taylor and Hutton to introduce the President-elect, George Westinghouse.

Mr. Westinghouse then took the chair.

There were present at the meeting: President, George Westinghouse; Vice-Presidents, Chas. Whiting Baker, Geo. M. Bond, R. C. Carpenter, W. F. M. Goss, E. D. Meier, F. M. Whyte; Managers, J. Sellers Bancroft, H. L. Gantt, James Hartness, Alex. C. Humphreys, I. E. Moultrop, H. G. Reist, H. G. Stott; Past-Presidents, F. R. Hutton, Charles Wallace Hunt, Jesse M. Smith; Chairman Finance Committee, Arthur M. Waitt, and Secretary, Calvin W. Rice. Regrets were received from Treasurer, Wm. H. Wiley, and Manager, W. J. Sando.

The minutes of the meeting of December 7 were read and approved.

In the absence from the room of Calvin W. Rice, Secretary of the Society for the year 1909, H. G. Stott acted as Secretary *pro tem.*

Voted: That Calvin W. Rice be elected Secretary for the year 1910 on the same terms as the previous year.

Voted: That F. R. Hutton be elected Honorary Secretary for the year 1910, on the same terms as the previous year.

Voted: That Jesse M. Smith, Past-President, be elected Trustee of the United Engineering Society to serve for a term of three years, to fill the vacancy created by the expiration of the term of office of Charles Wallace Hunt.

Voted: That Henry R. Towne be reappointed a member of the John Fritz Medal Committee, under the provisions of C-46 and B-32 to serve for a term of four years, to succeed himself.

Voted: That the Council delegate to the President the appointment of the Executive Committee of the Council for the year 1910 and until the appointment of the new Executive Committee the present Executive Committee continue in service.

The meeting adjourned to January 11, 1910.

THE NEWLY ELECTED OFFICERS FOR 1910

GEORGE WESTINGHOUSE

President Am. Soc. M. E.

George Westinghouse, a son of George and Emeline Vedder Westinghouse, was born at Central Bridge, N. Y., October 6, 1846. His father was a manufacturer of agricultural machinery, and established works at Schenectady, which are still in operation. The younger Westinghouse was educated in the public schools and at Union College, Schenectady, and received his early mechanical training in his father's manufactory. His tastes were strongly in the direction of machinery and the solution of mechanical problems.

The patriotic ardor which filled the youth of the country during the civil war drew young Westinghouse into the volunteer army in June 1863. He was under seventeen, but on account of his size and strength—he was six feet tall and weighed 180 lb.—the recruiting officers admitted him without asking his age. He enlisted with the Twelfth New York National Guard. Subsequently, he joined the Sixteenth New York Cavalry, and in December 1864 became an assistant engineer in the United States Navy, serving in that capacity until August 1865.

Returning to civil life he invented in the same year a device for replacing derailed cars, and while placing this invention with the railroads his attention was attracted by the prevalence of minor and serious accidents due to the lack of efficient means for controlling trains in motion. After a careful study of the subject, and such experiments as were possible with the limited means then obtainable, he invented the air brake and patented it in 1868.

The first train to which this brake was applied ran on a line west from Pittsburgh and on what is now a portion of the Pennsylvania Railroad. During the trial trip a collision with a loaded team stuck on a grade-crossing was prevented. This practical illustration of the utility of the invention led to the adoption of the brake. Mr. Westinghouse, retaining the control of his invention, undertook to manufacture it and organized the Westinghouse Air Brake Company,

establishing at Pittsburgh the business which subsequently became the nucleus of the many industries associated with his name.

From the invention of the air brake dates the beginning of modern railroading. The air brake is primarily a train-operating device which makes possible the fast and long trains, large cars, heavy loads and frequency of service of the present day, and the numerous improvements which Mr. Westinghouse has wrought in his invention have kept its efficiency well in advance of the new and varied conditions which constantly arise. Before he was twenty-five his name had become familiar throughout the world, and his contribution to the material progress of civilization was everywhere recognized. He continued in the study and practice of engineering, and equipped a machine shop for his personal experimental use, where he worked out many inventions, at first relating almost entirely to devices for railroad operations. He applied compressed air to switching and signalling and later utilized electricity in this connection. From this grew the Union Switch and Signal Co.

His introduction of electricity into switch and signal work led him far into electrical experiment and he devoted his energies to a cause in which few then believed, the adoption of the alternating current for lighting and power, in which he had to meet and overcome almost fanatical opposition, which in many States sought legislation against the use of the alternating current as dangerous to the public welfare. In 1885 he acquired the patents of Gaulard & Gibbs, and having undertaken a comprehensive study of the distribution and utilization of electrical currents in a large way, he personally devised apparatus and methods for the work, and gathered around him a group of men who were to become experts in the new electrical art. He also organized the electrical company which bears his name and undertook the development and manufacture of the induction motor which made practical the utilization of the alternating current for power purposes.

Following the discovery of natural gas in the Pittsburgh region, Mr. Westinghouse devised a system for controlling the flow and for conveying the gas over long distances through pipe lines, thus supplying fuel to the homes and factories of Pittsburgh. He took up the study of the gas engine, and for ten years conducted a series of exhaustive experiments in this line, at the end of that time putting into commercial use a gas engine of large power for electric generating.

Mr. Westinghouse introduced the Parsons steam-turbine into this country, adding to it improvements and developments of his own,

and others carried out under his supervision. He also has recently developed a steam turbine for ship-propulsion designed to overcome the well-known objections to the use of turbines in that field, and lately coöperated with Rear-Admiral Melville and John H. Macalpine in their study of problems associated with driving of propellers at low speed by turbines of high speed.

It is impracticable to enumerate here the inventions which Mr. Westinghouse has personally made or those which his staff have brought forth under his supervision. As a result of this work and enterprise, there have grown thirty corporations of which he is president, employing 50,000 men, $120,000,000 of capital, with works at Wilmerding, East Pittsburgh, Swissvale and Trafford City, Pa.; at Hamilton, Canada; London and Manchester, England; Havre, France Vardo, Italy; and at Vienna and St. Petersburg.

Mr. Westinghouse has made many visits to Europe in connection with his inventions and industries. There as in his own country he has won the friendship of the foremost men of his time and the high esteem of the engineering profession. He has been decorated by the French Republic and by the sovereigns of Italy and Belgium; and he was the second recipient of the John Fritz Medal, Lord Kelvin, his friend of many years, having been the first. The Königliche Technische Hochschule of Berlin bestowed upon him the degree of Doctor of Engineering; and his own college, Union, gave him the degree of Ph.D. In 1905 Mr. Westinghouse was selected as one of the three trustees in whose hands the voting power of the controlling stock interest in the Equitable Life Insurance Society was placed. The other trustees were Ex-President Grover Cleveland, and Justice Morgan J. O'Brien. The selection of these three men met with universal approbation. Besides his Honorary membership in The American Society of Mechanical Engineers, Mr. Westinghouse is one of the two honorary members of the American Association for the Advancement of Science and is an honorary member of the National Electric Light Association.

Mr. Westinghouse married, in 1867, Miss Marguerite Erskine Walker and has one son, George Westinghouse, Jr. While he claims Pittsburgh as his residence, he has also a country home at Erskine Park, Lenox, Mass., as well as a house in Washington.

VICE-PRESIDENTS

CHARLES WHITING BAKER

Charles Whiting Baker, editor and vice-president of Engineering News, was born in Johnson, Vt., January 17, 1865, and was educated at the State Normal School at Johnson and at the University of Vermont. He received the degree of Civil Engineer from the latter institution in 1886. During his course Mr. Baker spent one vacation as aid on triangulation work for the United States Coast and Geodetic Survey in Vermont, and on graduation he worked for a few months in the drafting room of the Baldwin Locomotive Works, at Philadelphia, Pa.

Leaving this position in February 1889 to become associate editor of Engineering News, of New York, Mr. Baker took up a work which has claimed his attention ever since. Since 1892 he has been in practical charge of the editorial department, becoming in 1895, on the death of A. M. Wellington, managing editor and secretary of the company. Ten years later he became vice-president.

Mr. Baker published in 1889 an economic work, Monopolies and the People, and he has contributed to the Society a paper entitled, What is the Heating Surface of a Steam Boiler, presented in June 1898. He joined the Society in 1893, and served on the Meetings Committee from 1905 to 1908.

WILLIAM FREEMAN MYRICK GOSS

William Freeman Myrick Goss was born in Barnstable, Mass., October 7, 1859. In 1879 he received the certificate of the Massachusetts Institute of Technology, and afterwards the degree of Hon. M.S., from Wabash in 1888, and D.Eng., from the University of Illinois in 1904.

In 1879 Dr. Goss became an instructor in the department of mechanic arts of Purdue University, and remained in the service of that institution for nearly thirty years, becoming successively professor of practical mechanics in 1883 and professor of experimental engineering in 1889. He was made a director of the engineering laboratory in 1899, and dean of the school of engineering in 1900. In 1907 Dr. Goss entered the University of Illinois as dean of the college of engineering and director of the school of railway engineering and administration.

Dr. Goss served on the jury of awards for the Columbian Exposition in 1893, and has been a member since 1906 of the executive committee of the National Advisory Board on Fuels and Structural Materials. He is a fellow of the American Association for the Advancement of Science, and a member of the Society for the Promotion of Engineering Education, the Western Railway Club, the Western Society of Engineers, the American Institute of Electrical Engineers, the Illinois Academy of Science, the Master Car Builders' Association, the Master Mechanics' Association, the International Association for Testing Materials, and the Illinois Society of Engineers and Surveyors.

Dr. Goss has made a specialty of the subject of steam engineering, investigating largely the economic performance of locomotives, high pressure in locomotive service, superheated steam in locomotive service, behavior of car axles, friction brakes, front-end arrangement of locomotives, fuel briquets in locomotive service, power transmission by friction wheels, graphite as a lubricant, etc.

Dr. Goss is a life member of this Society, which he entered in 1886. He was a member of the board of managers from 1900 to 1903, and has served on many committees. He has contributed the following papers: The Cole Locomotive Superheater; A Series Distilling Apparatus of High Efficiency; The Effect of the Counterbalance in Locomotive Drive Wheels upon the Pressure between Wheel and Rail; Tests of a Ten-Horsepower DeLaval Steam Turbine; New Forms of Friction Brakes; Tests of the Locomotive at the Laboratory of Purdue University; Paper Friction Wheels; Tests of a Twelve-Horsepower Gas Engine; Efficiency Tests of a One Hundred Twenty-Five Horsepower Gas Engine; The Effect upon the Diagrams, of Long Pipe Connections for Steam-Engine Indicators; Test of the Snow Pumping Engine at the Riverside Station of the Indianapolis Water Company; Locomotive Testing Plants; Power Transmission by Friction Driving; The Conservation of the Nation's Fuel Supply; The Debt of Modern Civilization to the Steam Engine.

EDWARD DANIEL MEIER

Colonel Edward Daniel Meier, president and chief engineer of the Heine Safety Boiler Conpamy, was born in St. Louis, Mo., May 30, 1841. At the close of a scientific course at Washington University, St. Louis, he studied four years at the Royal Polytechnic College at Hanover, from 1859 to 1862. He was then apprenticed to Wm. Mason's

Locomotive Works at Taunton, N. J. He left this company for military service, part of the time doing construction work as assistant engineer on the defenses of New Orleans.

In 1865 Colonel Meier entered the Rogers Locomotive Works at Paterson, N. J., as machinist and draftsman, During the next ten years he held various important positions with the Kansas Pacific Railway, the Illinois Patent Coke Company, the Meier Iron Company, the St. Louis Interstate Fair, and the St. Louis Cotton Factory. From 1876 to 1879 he was designer and superintendent of the Peper Hydraulic Cotton Press, and after two years of varied administrative work became president and chief engineer of the Heine Safety Boiler Company, which offices he still holds. During that period he has acted as consulting engineer on the Union Depot Railway of St. Louis, constructing the first electric power station in that city; and from 1902 to 1908 as engineer-in-chief and treasurer of the American Diesel Engine Company.

Colonel Meier has held office in the St. Louis Engineers' Club, the American Boiler Manufacturers' Association, and the Machinery and Metal Trades Association. He entered this Society in 1891 and has served it as manager, from 1895 to 1898, and as vice-president from 1898 to 1900.

MANAGERS

J. SELLERS BANCROFT

Mr. J. Sellers Bancroft was born September 12, 1843, and was educated in the public schools of Philadelphia.

In March 1861 he was apprenticed to the machinery business with Wm. Sellers & Co., with whom he was advanced to gang foreman in 1863, before the completion of his apprenticeship, and shop foreman in 1867, becoming a member of the firm in 1873, and manager of the business from its incorporation in 1887 to January 31, 1902, when he left this company to become general manager and mechanical engineer for the Lanston Monotype Machine Company, builders of monotype-casting and composing machinery. Mr. Bancroft has taken out over sixty patents for various inventions in machine tools, injectors, testing machines, electrical appliances, and type-casting and composing machines, and is largely responsible for the present condition of monotype machinery. He received a gold medal from the Paris Exposition of 1889 for his inventions in machine tools and injectors there shown.

Mr. Bancroft is a member of the American Association for the Advancement of Science, and has been a member of the Franklin Institute for over forty years. He entered this Society in 1880.

JAMES HARTNESS

James Hartness was born in Schenectady, N. Y., September 3, 1861, and received his early training in the public schools. After seven years of experience as machinist, toolmaker and draftsman, he became foreman and designer for the Union Hardware Company, of Torrington, Conn., a position which he relinquished after three years to become superintendent and designer for the Jones & Lamson Machine Co., of Springfield, Vt. He has had an active part in the management of this firm for nearly twenty-one years, and has been its president for the last nine years. Mr. Hartness is also president of the Bryant Chucking Grinder Company, treasurer of the Jones & Lamson Power Co., and director in a number of other machine tool building companies.

He has taken out seventy United States patents, besides many pending, his line of invention being machines for metal turning, notably the flat turret lathe.

Mr. Hartness published in 1909 a work on Machine Building for Profit and the Flat Turret Lathe, and has contributed to the Society, papers on Lead-Controlling Screw-Cutting Dies, and Tandem Dies, in 1897, and Metal-Cutting Tools without Clearance, in 1908. He joined the Society in 1891 and is a life member. He is also a member of the Institution of Mechanical Engineers of Great Britain, the American Society for the Advancement of Science, the American Institute for Scientific Research, and the Boston Chamber of Commerce, as well as the Engineers' Club, and various other social organizations.

HENRY G. REIST

Henry G. Reist was born near Mt. Joy, Lancaster County, Pa., May 27, 1862. He received from Lehigh University in 1886 the degree of M.E. The same year he entered the foundry and machine department of the Harrisburg Car Company. After a year of testing and erecting steam engines he became assistant superintendent of the company.

Leaving in the spring of 1889 to join the engineering excursion to Europe, he became associated on his return with the Thomson-

Houston Electric Company, at Lynn, Mass., having charge of the construction and testing of a large number of direct and alternating-current dynamos and stationary and railway motors. Soon after the consolidation of the Thomson-Houston Company with the General Electric Company, he took charge for them of the design of alternating-current generators and motors. When he was first engaged in electrical work, the largest machine manufactured by the company with which he was associated, was of 100-kw. capacity; now 14,000-kw. generators are regularly produced by this company.

Mr. Reist entered this Society in 1889 and somewhat later became a member of the American Institute of Electrical Engineers. He has contributed to the Society a paper on Blueprinting by Electric Light, and has presented papers before the American Institute of Electrical Engineers, and the Ohio Electric Lighting Association of Engine Builders, as well as a number of lectures to engineering students.

GENERAL NOTES

STUDENT BRANCHES

The following reports have come to the Society concerning the activities of its Student Branches:

At Columbia University the following officers were elected recently: F. R. Davis, president, H. B. Egbert, vice-president, H. B. Jenkins, secretary, and F. T. Lacy, treasurer. Papers are read before the organization once a month.

At Brooklyn Polytechnic a number of new members were received at the meeting of December 4, and a lecture was delivered by H. A. Black, on Depreciation Principles and Methods.

The Mechanical Engineering Society of the Massachusetts Institute of Technology enjoyed a lecture on December 21 by Robert A. Shailer, on Tunnels and Tunnel Construction. The society conducts excursions from time to time to places of industrial interest, those lately visited being the Quincy Market Coal Storage and Warehouse Co.'s refrigerating plant and the factory of the Stanley Motor Carriage Company. The officers are Frederick A. Dewey, chairman, Donald V. Williamson, vice-chairman, Arthur P. Truette, secretary, and Luke E. Sawyer, treasurer.

On December 3, the recently organized branch at the University of Cincinnati elected as temporary officers H. B. Cook, chairman, and P. G. Haines, secretary.

The Club of Mechanical Engineering of the University of Missouri, which was admitted at the last meeting of the Council on December 7 as a student branch of the Society, elected R. E. Dudley, president, Ernest C. Phillips, secretary-treasurer, and for members of the advisory board, Prof. E. A. Fessenden, Jun., Am.Soc.M.E., E. C. Phillips, and F. B. Thatcher. The club has as its honorary chairman Prof. Harry Wade Hibbard, Mem. Am.Soc.M.E.

Further statistics concerning these and other student branches are published on another page of The Journal.

ENGINEERS' CLUB BANQUET

The third annual banquet of the Engineers' Club took place Wednesday evening, December 22, with Mr. Andrew Carnegie, Honorary

Member Am.Soc.M.E., as the guest of honor. Mr. Carnegie mentioned his great pleasure in attending the dinner, an attendance which he regarded in the light of an obligation, and spoke again of his debt to the engineers and the chemists for their part in all his industrial success. He also repeated his prophecy that in the course of time Canada and the United States would be one nation.

This remark served to introduce another guest of the evening, Robert Cooper Smith, Esq. K.C., of Montreal, Quebec. Mr. Smith's address was an eloquent tribute to Mr. Carnegie. Mr. Martin W. Littleton followed and, in the absence of Hon. E. H. Gary, the speeches of the evening were concluded by Dr. Alex. C. Humphreys. Dr. Humphreys referred to the value of industrial education such as Mr. Carnegie is so successfully providing in the Carnegie Technical Schools at Pittsburgh, and reiterated his opinion that the educational work in America is too much influenced by the college, instead of training the average person for industrial life.

The attendance was nearly 200 and the excellence of the speeches and of the music, rendered by an orchestra under the direction of Hans Kronold, and the perfection of the menu and service made the occasion one of the most enjoyable and successful ever held by the Club.

DONATION TO THE LIBRARY

Clarence E. Kinne, Life Member, Am.Soc.M.E., in response to a request sent out through The Journal, has made up from his own files and sent to the Society the copies for 1894-1895, complete with index, forming vol. 1 of Machinery, which the Society had been unable to obtain through the customary channels. The volume has been placed in the Library, completing our files of this magazine to date.

REPRESENTATION AT FUNERAL SERVICES OF HORACE SEE

The President appointed James M. Dodge, Past-President, Rear Admiral George W. Melville, Past-President and Honorary Member, Oberlin Smith, Past-President, Fred. W. Taylor. Past-President, and J. Sellers Bancroft, Kern Dodge and Edward I. H. Howell, Honorary Vice-Presidents to represent the Society at the funeral services of Horace See, Past-President, Am.Soc.M.E., Thursday, December 16, 1909, at St. Peter's Church, 4th and Pine Sts., Philadelphia, Pa.

OTHER SOCIETIES

AMERICAN EXPOSITION IN BERLIN

As the first all-American Exposition ever conducted in a foreign country, the exposition to be held in Berlin during the summer of 1910 will offer peculiar advantages to American manufacturers. A freight reduction of 30 per cent both ways, granted by the Hamburg-American and the North German Lloyd lines, the remission of customs duty by the German Government, and the existence of the German-American patent treaty, which relieves American inventors from the necessity of obtaining patents in Germany, are among the inducements offered to exhibitors. The date set for the opening is June 20, and the exposition will be in progress three months. The exhibits will be carefully classified, the present plans including sections to be devoted to inventions, transportation, social economy and industrial safety, agricultural implements, machinery of all kinds, etc. Germany in 1908 consumed American products to the amount of $276,922,089; to say nothing of her influence on the trade of Europe.

The American headquarters for the exposition are in the Hudson Terminal Building, 50 Church St., New York, James L. Farmer, General Secretary. Members of the Society acting on the advisory committee are, C. A. Moore, Francis H. Stillman, Ambrose Swasey; and on the general committee, James M. Dodge and Thomas A. Edison, Honorary Member.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS

At a meeting of the American Institute of Electrical Engineers held in the auditorium of the Engineering Societies' building, December 16, a paper entitled Comments on Development and Operation of Hydroelectric Plants, was presented by Henry L. Doherty, Member Am.Soc.M.E. The meeting was under the auspices of the High-Tension Transmission Committee.

WESTERN SOCIETY OF ENGINEERS

The Western Society of Engineers has appointed the following as a committee to confer with the Chicago City Council and the Harbor Commissioner regarding harbor improvement and development: A. Bement, Mem.Am.Soc.M.E., *Chairman*, W. L. Abbott, Member of Council, Am.Soc.M.E., L. E. Ritter, E. C. Shankland, Mem.Am.Soc.M.E., Willard A. Smith.

AMERICAN INSTITUTE OF CHEMICAL ENGINEERS

The annual meeting of the American Institute of Chemical Engineers was held at Philadelphia, Pa., December 8 to 10. The address of welcome was made by Mayor John E. Reyburn. The following papers were presented for discussion: Natural Draft Gas Producers and Gas Furnaces, Ernest Schmatolla; The Commercial Extraction of Grease and Oils, W. M. Booth; The Chemical Industries of America, Prof. Chas. E. Munroe; Multiple Effect Distillation, F. J. Wood, Mem. Am.Soc.M.E.; The Advantages of the Multiple Effect Distillation of Glycerine and Other Products, A. C. Langmuir; Reclaiming of Waste India Rubber, S. P. Sharples; Materials for Textile Chemical Machines, Fred. Dannerth; A Method for Smelting Iron Ore in the Electric Furnace, Edw. R. Taylor; Chemical Composition of Illinois Coal, and Heat Efficiency of Smokeless Combustion and Heat Absorbing Capacity of Boilers, A. Bement, Mem.Am.Soc.-M.E.

Excursions were made to the laboratories of the University of Pennsylvania and the Commercial Museum, the chemical works of Harrison Bros. & Co., the Torresdale Filtration Plant; the wool-degreasing plant of Erben, Harding & Co.; the Welsbach Light Company, the plant of the Camden Coke Company; the Trenton Potteries; the Hamilton Rubber Company; the Linoleum Works; the cement plant at Allentown, Pa.

NATIONAL SOCIETY FOR THE PROMOTION OF INDUSTRIAL EDUCATION

The National Society for the Promotion of Industrial Education held its third annual convention at Milwaukee, December 2-4. The convention was opened with a public banquet at the Hotel Pfister, at which James O. Davidson, Governor of Wisconsin, presided. Addresses on the Economic Value of Industrial Education were made by Charles Van Hise, President of the University of Wisconsin, George

Martin, former secretary of the Massachusetts Board of Education, and Alex. C. Humphreys, Manager Am.Soc.M.E., President of Stevens Institute of Technology. Mr. Humphreys spoke particularly of the improvidence and superficial character of our educational processes which have built up a system that has the college as its goal, whereas in reality the masses need industrial training. The many are being sacrificed to the few.

Public meetings were held on the remaining days, at which National Legislation, Corporation Schools, Evening Schools, Industrial Education at Home and Abroad, and Intermediate Industrial Schools, were considered and discussed. Among those who addressed the gatherings were Willet N. Hayes, Assistant Secretary of Agriculture, John L. Shearer, President Ohio Mechanics' Institute, Arthur L. Williston, Mem.Am.Soc.M.E., Director in Pratt Institute, Mrs. Anna Garlin Spencer, Society for Ethical Culture, and Edgar S. Barney, Superintendent Hebrew Technical School for Boys. An exhibition of trade school work was conducted throughout the convention, some thirty prominent industrial institutions being represented.

The object of the society is to bring to public attention and to provide opportunities for the study of industrial education, as well as to make available the results of experience and to promote the establishment of additional institutions. Its work is carried on through a general office in New York and through State branches and committees. The New York State Branch has as its president James F. McElroy, Mem.Am. Soc.M.E., and as its secretary Prof. Arthur L. Williston, Mem.Am.Soc.M.E.

NATIONAL COMMERCIAL GAS ASSOCIATION

The fourth annual meeting of the National Commercial Gas Association occupied Madison Square Garden from December 14 to 22. One of the greatest undertakings of the exhibition committee was the piping of the entire building, making possible the most extensive and successful gas and gas appliance exhibition ever held. Among papers presented were: The Future of Gas for Street Lighting, E. N. Wrightington; The Use of Gas for Industrial Purposes, Present and Future, S. T. Wilson; The Application of Architectural Designs to Gas Fixtures, L. F. Blyler; Theory of Combustion, T. O. Horton; Gas Engines in Competition with Central Station Electric and Isolated Steam Plants, W. W. Cummings, Mem.Am.Soc. M.E.; General Maintenance and Special Troubles, R. H. Thomas;

Water Heaters, G. W. Savage. On December 17 a joint meeting of the Association with the New York Section of the Illuminating Engineering Society was held.

SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS

The seventeenth annual meeting of the Society of Naval Architects and Marine Engineers was held at the Engineering Societies Building, New York on November 18 and 19. Among the papers presented for discussion were the following: The Foreign Trade Merchant Marine of the United States; Can it Be Revived? by G. W. Dickie, Mem.Am.Soc.M.E.; the Evolution of Screw Propulsion in the United States, by Chas. H. Cramp; The Effect of Parallel Middle Body upon Resistance, by D. W. Taylor; The Applications of Electricity to the Propulsion of Naval Vessels, by W. L. R. Emmet, Mem.Am.Soc.M.E.; The Strength of Water-tight Bulkheads, by Prof. William Hovgaard; The Design of Submarines, by M. F. Hay.

The officers elected are: president, Stevenson Taylor, Mem.Am. Soc.M.E.; vice-presidents, J. W. Miller, Rear-Adm. Geo. W. Melville, Hon. Mem.Am.Soc.M.E.; Members of Council, Wm. J. Baxter, Geo. W. Dickie, Mem.Am.Soc,M.E.,W. D. Forbes, Mem.Am.Soc. M.E., Andrew Fletcher, Mem.Am.Soc.M.E., H. A. Magoun, Mem. Am.Soc.M.E., Lewis Nixon; Associate Members of Council, J. S. Hyde, Assoc.Am.Soc.M.E., C. B. Orcutt.

ENGINEERS' CLUB OF ST. LOUIS

At the annual meeting of the Engineers' Club of St. Louis, held in the club rooms on December 1, 1909, reports covering the work of the year were presented, and the following officers placed in nomination and ordered to ballot: President, M. L. Holman, Past-President, Am.Soc.M.E.; vice-president, J. D. Von Maur; secretary-librarian, A. S. Langsdorf; treasurer, C. M. Talbert; directors, J. W. Woermann and H. J. Pfeifer; members of the board of managers, Association of Engineering Societies, John Hunter, Montgomery Schuyler, J. F. Bratney.

The business of the evening was followed by an illustrated address on Reinforced-Concrete Construction by A. J. Widmer, of the Trussed Concrete Steel Company.

BROOKLYN ENGINEERS' CLUB

The Brooklyn Engineers' Club held its annual meeting in the clubhouse, 117 Remsen St., on December 9. The annual reports of the various committees and the Board of Directors were read. Upon motion it was voted that the election of the new board of officers take place during the annual dinner, Thursday, December 16, at which time the following were elected: President, George A. Orrok, Mem.Am.Soc.M.E.; secretary, Joseph Strachan; treasurer, William T. Donnelly, Mem.Am.Soc.M.E.; directors, William Andrews, Frederick C. Noble; auditing committee for one year, Fred. L. Cranford, Jacob Schmitt, Geo. A. Hartung. During the afternoon and evening of the same day, the first annual loan exhibition of the scientific books of the year, photographs and plans of engineering work was opened.

NECROLOGY

HORACE SEE, PAST-PRESIDENT, AM.SOC.M.E.

The sudden death of Horace See, Past-President, Am.Soc.M.E., December 14, 1909, is announced. An account of his life will appear in an early number of The Journal.

The death of Dr. Charles B. Dudley, member of the Research Committee of the Society, December 21, 1909, is announced. An account of his life will appear later.

CHARLES HENRY WILLCOX

Charles Henry Willcox died at his home in Westport, Conn., on September 13, 1909. Mr. Willcox was born in Little Falls, N. Y., on March 31, 1839, and was the son of James Willcox, founder and president of the Willcox & Gibbs Sewing Machine Co. He entered his father's business at the age of eighteen and was continuously connected with the company as mechanical engineer from 1866 until his retirement a few years ago, and most of that time as director. The natural bent of his mind was toward mechanics and in collaboration with James E. A. Gibbs he developed and placed on the market the invention of the single-thread chain-stitch sewing machine, which is now so widely used in the making of wearing apparel. Other patents followed, in particular that of the automatic tension, which it is said consumed ten years of patient experimentation before it was perfected. Mr. Willcox was also the inventor of two straw-hat sewing-machines, one the American straw hat machine in which the stitch is visible, and another in which the stitch is concealed. These two machines are used to-day in the manufacture of fully 90 per cent of all straw hats made. The knit goods manufacturing field also received an impetus through the invention of the Willcox & Gibbs hosiery trimming machine. The overlock machine worked out by Mr. Willcox in collaboration with the late Stockton Borton, was a great advance over the hosiery trimming machine and is recognized as one of the finest mechanical productions

in sewing machines. Through the ornamental character of its stitch it has been adopted in lines of manufacture other than that for which it was originally intended.

In addition to his connection with the Willcox & Gibbs Sewing Machine Co., Mr. Willcox was for forty years affiliated with the Brown & Sharpe Mfg. Co. He was a life member of the Society.

CHARLES SWINSCOE

Charles Swinscoe, consulting engineer of the Clinton Wire Cloth Company, Clinton, Mass., was born at Nottingham, England, January 1, 1833. His early education was received in the Collegiate School, Manchester, England. He came to this country when a lad and at one time was Fourth Officer on the Dreadnought under Capt. Samuel Samuels.

From 1851 to 1854 Mr. Swinscoe studied practical mechanics in his father's shop. In 1867 he established the steam pump works of the Geo. F. Blake Mfg. Co., at Boston, designing most of the work. In 1876 he left this company to take charge of the Reading Hydraulic Works, designing its steam pumping machinery. From 1878 to 1880 he was in charge of the Bay State Brick Company and after that date of the Clinton Wire Cloth Company. In 1903 he became consulting engineer of this company.

Mr. Swinscoe was a musician of ability and was president of the Clinton Choral Union and organist of the Episcopal Church for many years. He was a member of the Clinton Historical Society, and a member of this Society since 1887.

PERSONALS OF THE MEMBERSHIP, AM. SOC. M. E.

Ludwell B. Alexander has assumed the position of vice-president of the Haggerty Contracting Company, Bronx Borough, New York. He was formerly associated with the United Engineering and Constructing Company, New York, as assistant engineer.

Thomas Appleton, formerly connected with the East St. Louis, Ill., office of the U. S. Public Buildings, as superintendent of construction, is now identified with the Alton, Ill., office.

Adolph O. Austin, chief draftsman of the Starr Engineering Co., New York, has accepted a position with the Vilter Mfg. Co., Milwaukee, Wis., in the capacity of assistant engineer.

C. Kemble Baldwin, formerly chief engineer of the Robins Conveying Belt Company, and for the past two years chief engineer of the Robins New Conveyor Company, has been appointed chief engineer of the Robins Conveying Belt Company, the two companies having been consolidated. Mr. Kemble lectured on The Belt Conveyor, on November 10, before 400 members of the first class of the engineering course of the University of Illinois.

A. Bement presented papers on Chemical Composition of Illinois Coal, and Heat Efficiency of Smokeless Combustion and Heat Absorbing Capacity of Boilers, at the December 8–10 convention of the American Institute of Chemical Engineers, held in Philadelphia, Pa.

Paul P. Bird presented a paper on The Smoke Problem of Chicago at the November 17 meeting of the Western Society of Engineers.

Walter J. Bitterlich, formerly machine designer with the Bresnahan Shoe Machinery Company, Lynn, Mass., has accepted a position with the Hood Rubber Company, Watertown, Mass., to act in the capacity of chief draftsman.

Paul M. Chamberlain has resigned the position of chief engineer of the Underfeed Stoker Company of America, Chicago, Ill., to take up private practice. His office will be in the Marquette building, Chicago, Ill.

Chas. C. Christensen contributed an article on A One Hundred Ton Modern Cyanide Plant to the November 13 issue of *The Mining World*.

H. V. Conrad has accepted a position with the Westinghouse Air Brake Company, Wilmerding, Pa.

George L. Crook, recently in charge of the manufacturing organization in the E-M-F plant at Detroit, Mich., has entered the employ of the M. Rumely Co., La Porte, Ind., as works manager.

Henry L. Doherty presented a paper entitled, Comments on the Development and Operation of Hydro-Electric Plants, at the December 16 meeting of the American Institute of Electrical Engineers.

Carl S. Dow contributed an article on The Fuel Economizer to the December issue of *The Practical Engineer.*

Frank B. Gilbreth is the author of a book on Bricklaying System.

Charles A. Hague delivered a lecture on The Development of the Pumping Engine, at the Sheffield Scientific School, Yale University, New Haven, Conn., November 12.

An article on Errors in Grinding Tapered Reamers and Milling Cutters, by H. A. S. Howarth, was published in the December number of *Machinery.*

Prof. Fred. R. Hutton delivered a lecture, on November 9, on Some Problems of the Large Gas Engine, before the Stevens Institute Engineering Society, affiliated with The American Society of Mechanical Engineers. Professor Hutton has been invited to give a lecture before the Graduate School of Marine Engineering, U. S. Naval Academy, Annapolis, in January.

A. Lewis Jenkins has contributed an article on Stresses due to Bending and Twisting and the Design of Shafting, to the November 12 issue of *Engineering* (London).

Charles Kirchhoff, who has been connected for almost thirty years with *The Iron Age,* and the other publications of the David Williams Company, has disposed of his interests in that company and retired from active business.

George L. Knight delivered a lecture on The Generating and Distributing System of the Brooklyn Edison Company, at the November 6 meeting of the Brooklyn Polytechnic Student Branch of the Society.

Prof. A. G. Koenig delivered an illustrated lecture on Refrigeration before the December 7 meeting of the Modern Science Club.

J. W. Lieb, Jr., delivered a lecture, December 10, before the Electrical Engineering Society of Columbia University, the subject being Electric Light.

John McGeorge and H. W. Woodward have formed a consulting firm under the name of the Cleveland Engineering Company, with offices in the New England Building, Cleveland. Mr. McGeorge has been chief engineer of the Wellman-Seaver-Morgan Co., Cleveland, O.

A biographical sketch of Spencer Miller was published in the December issue of *Cassier's Magazine.*

David M. Myers contributed an article on Burning Natural Gas as Boiler Fuel to the December 7 issue of *Power and the Engineer*.

E. W. Nicklin, recently identified with the Diamond Power Specialty Company, Detroit, Mich., has accepted a position with the Detroit Brass Works, Detroit, Mich.

George A. Orrok lectured before the Student Section of The American Society of Mechanical Engineers at Columbia University on the evening of December 3 on Gas Engine and Blast Furnace Practice. At the December 21 meeting of the Modern Science Club, Mr. Orrok delivered a lecture on Surface Condensers. Mr. Orrok has been elected president of the Brooklyn Engineers' Club.

Thos. C. Pulman, formerly manager in India for the Worthington Pump Company, Ltd., and James Simpson & Co., Ltd., subsidiary companies of the International Pump Co., of New York, has been appointed to the London offices of the companies, to supervise the Indian and Eastern business, and will be attached to the sales department.

R. H. Rice addressed a joint meeting of the Electrical Section of the Western Society of Engineers and the Chicago Branch of the American Institute of Electrical Engineers, December 22, on Low Tension Feeder Systems for Street Railways.

Morris DeF. Sample, formerly manager of department, National Patent Holding Company, Chicago, Ill., has become associated with The Fire Protection Company, Indianapolis, Ind., as secretary-treasurer.

Charles M. Schwab has been elected a trustee of Lehigh University.

O. G. Smith, associated with the Platt Iron Works Company, Dayton, O., has been made manager of the company's branch house at St. Louis, Mo.

Arthur C. Tagge, formerly identified with the Eastern Canada Portland Cement Co., Dombourg, P. Q., has become associated with the Canada Cement Co., Montreal, P. Q.

Stevenson Taylor has been elected president of the Society of Naval Architects and Marine Engineers.

Edward P. Thompson, formerly of New York, has moved his business to Washington, D. C., in order to be near the Patent Office in behalf of clients.

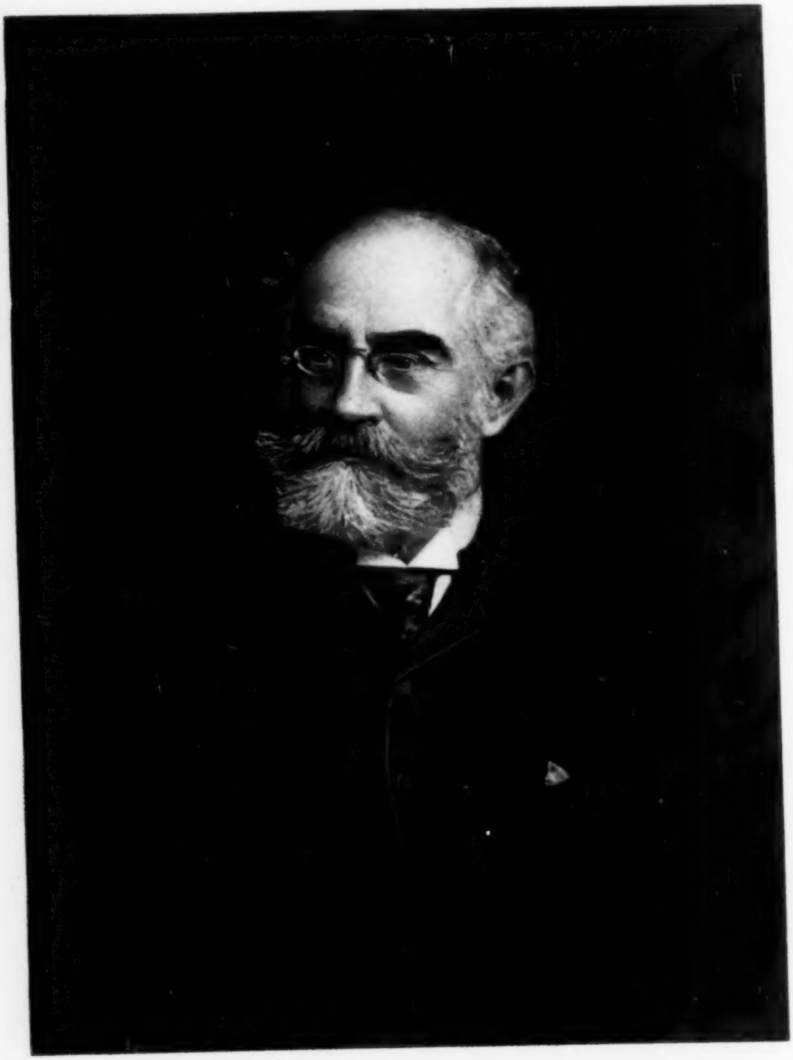
S. K. Thompson, formerly master mechanic with Stanley G. Flagg & Co., Philadelphia, Pa., has established an office in the Real Estate Trust Building in that city as consulting mechanical engineer.

S. Tompkins, who has been in charge of the shops and engineering department of the Miller School in Virginia, has been appointed superintendent of power

stations and chief engineer of shops and track, of the Coney Island & Brooklyn R. R., Brooklyn, N. Y.

Walter H. Trask, Jr., has been appointed district sales manager of the Denver Engineering Works Company, Salt Lake City. He was formerly assistant to sales manager in the company's main office in Denver.

F. J. Wood presented a paper on Multiple-Effect Distillation at the December 8 to 10 meeting of the American Institute of Chemical Engineers, held in Philadelphia, Pa.



PRESIDENT 1888

OF

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS

THE ELECTRIFICATION OF TRUNK LINES

By L. R. Pomeroy, New York
Member of the Society

It is assumed from a physical and mechanical viewpoint, that electric traction can meet all the demands and requirements of railroad service. Therefore, whether electricity will replace steam traction or not is entirely a commercial problem.

THE COMMON DENOMINATOR—COMMERCIAL CONSIDERATIONS.[1]

2 It may be stated at the outset that whatever system of electrification is adopted, a very large outlay has to be faced and no case for electrification can be made out unless an increase in net receipts can be secured sufficient to more than pay interest on the extra capital involved. This increase may be brought about either by decreasing the working expenses for the same service, by so modifying the service as to bring in a greater revenue, or by a combination of these.

3 However, there is hardly a steam road in existence to-day which does not have divisions or sections, where distinctly local traffic can be handled more profitably by light, comparatively frequent electric service, than as now, with heavy steam trains. Both steam and electric service can be operated over the same tracks without detriment or embarrassment to either. In so doing each kind of

[1] Commenting on the problem of electrification of the Central Pacific over the Sierras, Mr. Kruttschnitt says: "Eastern critics may be inclined to the opinion that we are dallying with this matter. We have found that it pays well to make haste slowly with regard to innovations. Electrification for mountain traffic does not carry the same appeal that it did two years ago. Oil burning locomotives are solving the problem very satisfactorily. Each Mallet compound locomotive, having a horsepower in excess of 3,000, hauls as great a load as two of former types, burning 10 per cent less fuel and consuming 50 per cent less water."—*Wall Street Journal.*

The American Society of Mechanical Engineers.
All papers are subject to revision.

service would be appropriately handled in a manner best suited to the conditions of each.

4 The fundamental principle, based on the present state of the art, seems to be that if you cannot accomplish something by means of electricity that is now impossible by steam traction, there is nothing to justify the change; the mere substitution of one kind of power for another, merely to obtain the same result, is not commercially warranted.

5 There are certain inherent advantages in electrical operation that have shown up very well, because the increase in business has absorbed the increased interest account, but these cases hardly apply to trunk line conditions as the law of induced travel has no bearing on freight train operation, the principal business of trunk line roads.

6 In heavy work the limiting feature of the steam locomotive is the boiler, and the maximum adhesion can be utilized only at low speeds. For example, a 2-8-0 locomotive with 180,000 lb. on the drivers, has a tractive force, at 10 miles per hour, of about 40,000 lb. or 4.5 to 1. At 30 miles per hour the tractive force becomes 13,250 lb. or 30.2 to 1. As tractive force governs the tonnage hauled, the ability of the electric locomotive to utilize almost indefinitely power proportional to the maximum adhesion and produce a drawbar pull entirely independent of the critical speed of a steam locomotive, as limited by the boiler, is a marked feature.

7 In heavy grade work the ability to increase the speed shows up favorably to the electric locomotive as enlarging the capacity of a given section, but here also the business has to be sufficient to absorb the increase in fixed charges.

8 With steam locomotives a coal consumption, when running, of 4 to 5 lb. per i.h.p. hr. really means 6 or 7 lb. at the rail, when the losses due to firing up, laying by in yards and sidings, blowing off at the pops, and consumption of the air pumps, are taken into account. Whereas, under electric operation, with an efficiency of 65 to 70 per cent between the power house and the rail, a coal consumption of 4 lb. per kilowatt hour at the rail can be counted on.

9 The writer is informed that the Metropolitan Street Railway station (1903) with a 40 per cent load factor, produced power, at the switchboard, at the rate of 4.7 mills per kilowatt hour (or 3.5 mills per horsepower hour), and with a load factor of 55 per cent which prevails in the winter time, the cost is at the rate of 4.43 and 3.3 mills respectively. These costs cover all expenses and repairs except

fixed charges. The coal consumption is 2.9. lb. per kilowatt and 2.16 per horse-power hour.

10 L. B. Stillwell is authority for the statement that the Interborough is producing power at the rate of 2.6 lb. of coal per kilowatt hour or 3 lb. at the drawbar.

11 Another authority gives the following figures for the elevated roads for cost of power, \$0.005 per kilowatt hour at the switchboard, \$0.0066 at the third rail shoes, or \$0.0089 at the rims of the drivers. These figures are exceptional and hard to duplicate and as the fixed charges are not included, the writer would consider $1\frac{1}{4}$ cents per kilowatt hour at the rail a conservative figure, and will use this cost in the following computations.

RELATIVE COST OF COAL FOR STEAM AND ELECTRIC OPERATION

12 It may be fair to assume that where average coal is used, we can count on about \$2.25 per ton for locomotive coal on the tender, while a much cheaper grade can be used in the power house, costing, with modern coal handling facilities, about \$1.50 per ton. At this rate the relative difference in the cost of coal at the rail would be represented by the following figures:

Electric Power Station $\dfrac{2.5 \text{ lb} \ldots}{50\% \text{ off.}} \times \$1.50$	\$7.50
Steam Locomotive $7 \times \$2.25$	\$15.75

or 50 per cent in favor of electricity. The following results of the Mersey Tunnel operation are pertinent: Under electric operation one ton of coal at \$2.10 yields 2.29 ton miles at $22\frac{1}{2}$ miles per hour, while with steam, one ton of coal, at \$3.84 yields 2.21 ton miles at $17\frac{3}{4}$ miles per hour. The difference amounting to 55 per cent is in favor of the electric operation, thus:

$$\left[1 - \frac{2.10}{3.84}\right] \times \frac{22.5}{2.29} \div \frac{17.75}{2.21} = \left[1 - \frac{2.10}{3.84}\right] \times \frac{22.5 \times 2.21}{2.29 \times 17.75} = 55\%$$

13 On mountain grades or in heavy freight service, where the boiler of the freight locomotive is forced to the limit, and the boilers are designed for this particular purpose, the showing is still more favorable to the electric side. Especially is this true when the steam locomotive is detained on side tracks for as long a period as it takes to make the run, which is very frequently the case, since under these conditions the cost for fuel becomes a larger proportion of the

total operating expense. A 2–8–0 locomotive with 50 sq. ft. of grate surface burns 300 lb. of coal per hour while lying on side tracks. Reports from Mallet locomotives indicate that from 600 to 800 lb. are burned per hour under the same conditions.

14 The cost of a unit of power with the steam locomotive becomes relatively higher under maximum than minimum boiler demands, while with electricity the cost per unit is at a uniform rate, whether working under extreme or light power demands.

For example:

15 *Case 1.* A consolidation (2-8-0) type locomotive with 180,000 lb. on 57 in. drivers, 50 sq. ft. of grate surface, working under maximum conditions on a $1\frac{1}{2}$ per cent grade, would burn 150 lb. of coal per sq. ft., of grate surface per hour and evaporate from 12 to 15 lb. of water per sq. ft. of heating surface per hour. Under these conditions the cost per 1,000 ton miles would figure out as follows:

$$\frac{F \times \text{price per ton} \times R \times 1000}{2000 \times \text{m.p.h.} \times E \times TF} = \text{Cost per 1,000 ton miles}$$

where F = coal per hour (150 lb. $\times$ 50 sq. ft. of grate surface).
R = resistance to be overcome [(grade per cent $\times$ 20) plus 6].
E = 80 per cent efficiency to cover losses such as cleaning fires, idle time while under steam, cylinder condensation, air pump consumption, etc.
TF = tractive force, in this case 180,000 lb. on drivers $\div$ 4.5 = 40,000 lb.

Substituting these values, the formula becomes

$$\frac{7{,}500 \text{ lb.} \times \$2.85 \times 36 \times 1{,}000}{2{,}000 \times 10 \times 80\% \times 40{,}000} = \$1.20$$

If the same service is handled by electric locomotives the cost on a similar basis becomes:

$$\frac{R \times (\text{watt hr. per ton mile}) \times 1{,}000 \text{ tons} \times \text{price per kw. at the rail}}{1{,}000 \text{ watts}}$$

$$= \frac{36 \times 2 \times 1{,}000 \times \$0.01\frac{1}{4}}{1{,}000} = \$0.90$$

17 If locomotive coal is taken at \$1.70 per ton (the price in eastern Pennsylvania for low grade soft coal), the cost for coal for locomotives under the foregoing conditions would be:

$$(a) \quad \text{Steam, } \frac{\$1.20 \times 1.70}{2.85} = \$0.716$$

(b) Electric current reduced to 1c. per kw. hour at the rail:

$$\frac{0.90 \times 1\text{c.}}{1\frac{1}{4}\text{c}} = \$0.72$$

18 *Case 2.* An express passenger locomotive of the Atlantic (4-4-2) type, with the following data: Cylinders 21 by 26 in., boiler pressure 200 lb. per sq. in., weight on drivers 102,000 lb., heating surface 2,821 sq. ft., grate surface 50 sq. ft., rate of combustion 150 lb. per sq. ft. of grate surface per hour, speed 70 miles per hour. Figuring as in Case 1.

$$\frac{7{,}500 \times 2.85 \times 20 \times 1{,}000}{2{,}000 \times 70 \times 80\% \times 5{,}350} = \$0.71$$

Under electric conditions we have

$$\frac{20 \times 2 \times \$0.01\frac{1}{4} \times 1{,}000 \text{ tons}}{1{,}000 \text{ watts}} = \$0.50$$

or $28\frac{1}{2}$ per cent less.

19 If coal is taken at \$1.70 per ton, as in Case 1, the cost is reduced from \$0.71 to \$0.42, making the difference slightly in favor of steam.

20 These figures apply only to the conditions named, and average conditions on an undulating profile, when coasting is occasionally possible. With the benefits of momentum grades, also, the figures would be relatively less, but the electric locomotive would respond and benefit accordingly, so that the percentages would be approximately the same.

21 When steam locomotives are loaded to their capacity, as is generally the case where tonnage rating is practiced, the rate of combustion of 150 lb. of coal per square foot of grate surface per hour, will still hold good and remain constant, the tons hauled being the variable, responding or being modified by the speed or physical conditions of the road.

SAVINGS CLAIMED FOR ELECTRIFICATION

22 In view of the foregoing the following extract from an article by Mr. C. L. De Muralt will be of interest. The figures are from the annual report of 1903 of the roads named.

COST OF OPERATING TRUNK LINES

	P. R. R.	N. Y. C.
Fuel for locomotives	$6,000,135	$4,635,877
Water " "	335,286	295,583
Other supplies for locomotives	382,548	334,673
Wages: Engine men and roundhouse men	5,716,848	4,928,443
Other trainmen	4,442,127	2,991,335
Switchmen, flagmen and watchmen	3,900,427	2,511,552
Other expenses of conducting transportation	14,540,542	11,607,538
Repairs to locomotives	4,412,983	3,608,972
" other equipment	10,674,726	5,661,992
" roadbed	8,542,935	6,145,341
" structures	4,122,018	2,454,691
General expenses	1,858,319	1,786,494
	$64,928,894	$46,962,491

23 Mr. De Muralt then applies the figures found during the course of his investigation, which would lead to the following reductions if electricity was adopted as a motive power.

	P. R. R.	N. Y. C.
Fuel 10 per cent	$600,013	$463,388
Water saved entirely	335,286	295,583
Other supplies 50 per cent.	191,274	167,336
Wages, enginemen, etc., 25 per cent	1,429,212	1,207,361
Repairs to locomotives	2,206,492	1,804,486
Total amount saved	$4,762,277	$3,942,154

24 The saving in water alone capitalized at 5 per cent equals $6,750,000 for the former and nearly $6,000,000 for the latter road. As large as these alleged savings are, yet they would not amount to more than 2½ to 3 per cent on the necessary increase in capital to electrify the roads on which the foregoing savings apply.

25 While the first cost for power stations and electric equipment represents a large outlay, yet such items as the cost for repairs of locomotives and shops, expensive hostlering at terminals, coaling and water stations, and the incidental labor charge and repairs thereto will, in the aggregate, be materially reduced. The comparative saving in repairs will be indicated by the following figures:

Repairs	Steam	Electric
Boiler	20%	0%
Running gear	20%	20%
Machinery	30%	15%
Lagging and painting	12%	5%
Smoke box	5%	0
Tender	13%	0
	100%	40%

OTHER COMPARISONS BETWEEN STEAM AND ELECTRIC LOCOMOTIVES

26 It is further claimed that, with electric operation, greater mileage is possible with the electric locomotive and that fewer units are necessary to perform the same service. Great stress is laid on the fact that the ordinary freight locomotive makes only 3,000 miles per month, or 100 miles per day, against which is put forward

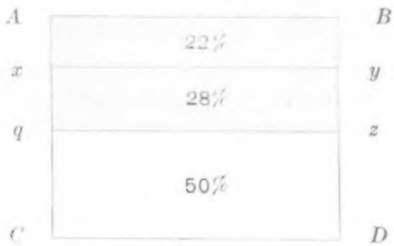


FIG. 1 DIAGRAM SHOWING DIVISIONS OF LOCOMOTIVE WORKING DAY

the ability of the electric locomotive to perform practically continuous service, suggesting the propriety of comparing electric and steam operation on the basis of ton miles per annum each is able to make and also the relative weight on driving wheels and not their total weight.

27 The operating efficiency of a steam locomotive in freight service is so low, averaging about 3,000 miles per month, that it is generally thought due to limitations, *per se*, in the locomotive, whereas it is mainly due to operating and traffic conditions, which limitations would apply with equal force to the electric locomotives, so that, barring some increase in speed, the electric locomotive can make no greater mileage than its steam competitor in equivalent service, consequently its splendid ability to perform almost continuous service cannot be realized in practice for reasons aforesaid.

28 Let the rectangle $A\ B\ C\ D$ represent a day of 24 hours the shaded area $A\ B\ x\ y$ that portion of the time for which the mechanical department is responsible = 22 per cent; the area $x\ y\ q\ z$, the average

time the locomotive is performing useful work = 28 per cent—*i. e.*, actually pulling trains, 3,000 miles per month, 100 miles per day; while the portion of the diagram bounded by $q\ z\ C\ D$, the period or balance of the time that the locomotive is under steam, with crew, and ready to go, and represents the time at terminal yards, side tracks and awaiting orders, etc. = 50 per cent.

29 It is just here that our electrical friends make the great mistake of claiming "greater capacity" for the electric locomotive over its steam equivalent. It is conceded that under electric conditions the area $A\ B\ x\ y$ may be reduced as much as one-half and perhaps, owing to greater speed, the area $x\ y\ q\ z$ may be increased, but the "lost motion" period due to traffic and operating causes will be relatively the same for both. The percentages are from an actual three months' test on a trunk line reported in1904 in the proceedings of the American Railway Master Mechanics Association by the committee on time service of locomotives.

30 The only cases where electric operation is commercially justified is in congested local passenger situations where the conditions closely approach those of a "moving sidewalk" and the records show that these cases have been profitable only when a large increase in business has been realized.

31 A modern Atlantic (4–4–2) type locomotive weighs, including tender, 321,620 lb. with a maximum tractive force of 23,500 lb. The ratio of total weight to tractive power is 133 to 1. The New York Central electric locomotive, with a total weight of 192,000 lb. and a tractive effort of 27,500 lb. has a ratio of 7 to 1. The comparison is still more favorable for electric freight locomotives where the entire weight is on the driving wheels.

POWER STATION CAPACITY

32 The impression is quite prevalent that if 100 steam locomotives are required to operate a certain division, if operated electrically, a power station capacity the equivalent of 100 locomotives would be necessary, whereas the generator capacity, barring the installation of spare units, would be of such size as to meet the average load. This average can be determined by laying down a train sheet, from which the load at any hour in the day can be seen and the peaks located.

33 For ordinary computations the number of trains to provide for is, approximately:

$$\frac{\text{The total train miles per hour}}{\text{Mean speed}}$$

This formula is the result of cancellation from the following:

(a) h. p. days ÷ Aggregate h. p.

That is:

$$(b)\ \frac{5{,}280 \times (\text{Dis. miles}) \times (\text{No. trains}) \times (\text{Tons}) \times \text{R}}{47{,}520{,}000 \text{ ft. lbs. in 1 day}}$$

$$\div \frac{\text{Tons} \times \text{R} \times \text{m.p.h.}}{375}$$

R = resistance due to gravity, + resistance due to speed, + curve resistance. Transposing and cancelling:

$$(c)\ \frac{\text{Dis. miles} \times \text{No. trains}}{24 \times \text{m.p.h.}}$$

For illustration take a typical case: Distance 183 miles.

LOAD		AVERAGE SPEED	
37 Freight Trains	at 15 m.p.h.	37 × 15 m.p.h. =	555
22 Expresses	at 50 m.p.h.	22 × 50 m.p.h. =	1,100
21 Locals	at 30 m.p.h.	21 × 30 m.p.h. =	630
80 Trains total.		80	2,285

2,285 ÷ 80 = 28 average m.p.h.

$$\frac{80 \text{ trains} \times 183 \text{ miles}}{24 \text{ hr.} \times 28 \text{ m.p.h.}} = 22 \text{ trains.}$$

34 For more accurate work a train sheet should be made either with miles as ordinates and time as abscissæ, or one with trains as ordinates on a time (abscissa) base.

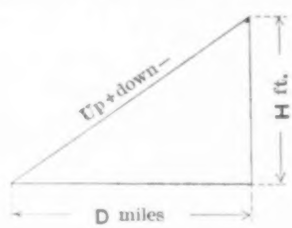
35 Relative to *R* (*i.e.*, resistance) for gravity; divide the profile into sections, one for each change in grade, plus or minus as the case may be:

$$\frac{\text{H}}{\text{D} \times 52.8} = \text{Per cent grade.}$$

Each 1% grade = 20 lb. = R

R for curves 0.56 lbs. per degree.

$$\text{R for level sections} = 2 + \frac{\text{m.p.h.}}{4}$$



36 Consider the example of a road or division 100 miles long on which a given train requires 2,000 h.p. to keep it in motion. If 20

cars take a maximum of 100 h.p. each, the electrical conductors and distributing apparatus will never be required to deliver more than 100 h.p. at any one point. If on the other hand, the entire traffic of the line must be concentrated in a single train, the electrical conductors and distributing apparatus must deliver the full 2,000 h.p. at each and every point. In other words, with the concentrated load, the capacity of the distributing apparatus at each and every point must be 20 times as great as the capacity when 20 cars are used to give the same total load. Electric traction has proved its superiority for distributing loads, but concentrated loads are still handled almost exclusively by steam locomotives.

SOME ADVANTAGES OF ELECTRIC LOCOMOTIVES

37 In the annual report of the P. R. R. (1903) the president states "That the congested condition of your system has brought about a large increase in the ton mile cost, which for 1903 was 25 per cent greater than for 1899. In order to prevent the increase in ton mile cost, it is necessary to move freight trains faster in places where traffic is dense, and for such purposes the electric locomotive is most efficient."

38 With steam locomotives the most economical average speed, for freight service, is 12 to 15 miles per hour, where there is ample track space for the free movement of trains. With a dense traffic this free movement can only be obtained by a higher speed and if the large train tonnage be maintained, more horse-power is required of the engine and boiler. It is difficult to increase the size of steam freight locomotives without resorting to the Mallet compound articulated type, and here we have the equivalent of two locomotives in one machine.

39 With the electric locomotive it is possible to develop a much greater horse-power and a large percentage of overload at the time when needed and do it more economically than with steam. The New York Central electric locomotive has a maximum peak horse-power of 3,000, which is 25 per cent above normal. This maximum is about double the power which can be obtained from the New York Central standard Atlantic (4–4–2) type locomotive. Similar proportions can be obtained for electric freight locomotives and their size and power are not limited by boiler capacity. If the steam locomotive is capable of developing 30,000 T. F. at the drawbar at 12 m.p.h or

$$\frac{30{,}000 \times 12 \text{ m.p.h.}}{375} = 960 \text{ h.p.}$$

and it is required to increase the speed of the train to 20 m.p.h. and maintain the same tonnage, then 1,600 horse-power will be required, which means the employment of a much larger locomotive or double heading.

40 The advantage of the overload capacity on short mountain grades or for strategic peaks is one of the strong points in favor of the electric machine and would make electric operation applicable to special cases rather than a universal substitute, in the broad light of commercial considerations.

GENERAL CONCLUSIONS

41 Our conclusion, from this survey of the situation, is that the rapid development of suburban passenger traction by electricity will require large power houses at large cities and these can gradually be made sufficient for working the line on further stretches in each direction, handling congested terminals, or used where commercially practicable, until it may be desirable to electrify the entire division.

42 Electric operation as compared with steam shows to greatest advantage in urban and suburban passenger service. Here, if multiple unit trains are employed, so that a considerable fraction of the total weight is carried on the driving wheels, thus permitting a high rate of acceleration to be used, a schedule speed quite impracticable in steam operation can be maintained. Moreover, a more frequent service can be given without a proportional increase in expense, whilst in times of light traffic small trains can be run, the energy consumption per train in such service being almost in proportion to the number of coaches. The law of induced travel, however, applies to urban and suburban passenger service, but does not hold for trunk lines and especially freight service.

TO DETERMINE WHETHER IMPROVEMENTS ARE JUSTIFIABLE

43 Under trunk line conditions the only thing that interests railway managers is the traffic available at the present, relatively speaking; the future is too indefinite to be capitalized to any great degree in advance. It is more in the line of insurance companies to "capitalize expectations."

44 In grade revision the authorization for expenditure is based on the saving in train miles capitalized. The following is a concrete case from a Western road, or rather the summation of the engineers'

report as to just what the proposed rearrangement would amount to. The rate of 50 cents per train mile is to cover those items of cost directly affected by the change.

$$\left.\begin{array}{c}\text{No. of}\\ \text{trains per}\\ \text{day—7}\end{array}\right\} \times \left[1 - \frac{\text{1,350 tons present conditions}}{\text{1,600 tons proposed}}\right]$$

$$\times \left\{\begin{array}{c}\text{Div. of}\\ 225\\ \text{Miles}\end{array}\right\} \times 50\text{c.} \times \left\{\begin{array}{c}365\\ \text{days}\end{array}\right\} = \$45{,}990.$$

45 Under the circumstances it will be seen that the value of 1 per cent reduction in train mileage per mile per train, amounts to $1.95 per annum. The total amount capitalized at 5 per cent equals $919,800. In some such manner the steam railroad manager arranges the proposition of the electric scheme and decides accordingly.

SOME EXAMPLES

46 In a paper before the American Society of Civil Engineers by W. J. Wilgus, some interesting data concerning New York Central operation were given:

Cost of coal per 2,000 lbs. anthracite steam loco., terminal service.......	$4.46
" " bituminous coal, road service..........	3.12
" " " " power station..........	2.72
Water per 1,000 gallons:—	
Power station..........	13.5 cts.
Road service..........	5 "

47 The cost of current, when power station designed load is attained, is 2.6 cents per kilowatt hour delivered at contact shoes. This includes all operating and maintenance costs, interest on the electrical investment required to produce and deliver current, depreciation, taxes, insurance and transmission losses. The following table summarizes the data:

Items	Operating Costs	Fixed Charges	Total
Power Station........	0.58 cts.	0.44 cts.	1.02 cts.
Transmission Losses	0.19 cts	0.15 cts.	0.34 cts.
Distribution Systems			
Substations.......	0.32 cts.	0.92 cts.	1.24 cts.
Totals............	1.09 cts.	1.51 cts.	2.60 cts.

48 In a discussion by G. R. Henderson (page 102, Vol. LXI, Trans. A. S. C. E.), are given road service costs per 1,000 car ton miles:

	Steam	Electric
Supplies	$2.03	$1.37
Wages	0.28	0.31
Interest, depreciation, and repairs to locomotive	0.46	0.34
	$2.77	$2.02

49 The item "Electric Supplies" is composed of operating expenses and fixed charges and may be analyzed thus:

53.3 kw. hour at $0.0109, $0.58 operation
52.3 kw. " " 0.0151, 0.79 fixed charges
52.3 kw. " " 0.026, 1.37

$$\left[\text{Fixed charges} = \left(\frac{0.79}{1.37}\right) = 57 \text{ per cent of operating expenses}\right]$$

The brackets are ours. The difference in cost between steam and electric traction in road service is $2.77 − 2.02 = $0.75 per 1,000 car ton miles.

50 The fixed charges on the power plant and the transmission system are $0.79 per 1,000 car ton miles, or about the same as the saving, so that if the train movement were but one-half the assumed amount (averaging 6,000 horse-power at the rails, or 6,000 kilowatts at the station) the cost for electric service would be slightly higher than for steam, or $2.81 as against $2.77 per 1,000 car ton miles.

51 The Manhattan Elevated, with about 38 miles of road, was electrified at an expense of $17,000,000. The operating ratio, under electric conditions, has been reduced from 61 to 46 per cent of gross receipts. The net result after taking care of the increased capital, etc., shows 15 per cent profit, but it is a significant fact that the increase in business was 46 per cent (carrying about 250,000,000 people per annum, 690,000 per day average, or 28,800 per hour).

52 There has just been reported the four years electric operating results of the Mersey tunnel road connecting Liverpool and Birkinhead. The net profit, allowing interest, etc., on the increased capital due to electrification, amounted to 15 per cent, but it took an increase in traffic of 55 per cent to make this operating result possible. Ton miles increased from 43 to 67 million, or 55 per cent. Total expenses, including interest on electric capital (but not depreciation) equal $0.586 per ton mile. Interest equals $0.106 per ton mile, or 22 per cent of operating expenses.

53 President Harahan of the Illinois Central reports the results

of the investigation that has been made relative to the proposed electrification in the following words:

54 "Our suburban traffic is the only service which would in any degree be adapted to electric operation, but even in this particular service it can be readily shown to be unjustifiable at the present time. I submit below a statement of the results which are estimated to accrue if the entire suburban service were electrified, compared with the present steam operation:

"Results of Operation of Suburban Business at Chicago for Fiscal Year ending June 30, 1909:

Gross earnings		$1,056,446
Operating expenses (82.9%) plus taxes		946,734
Net revenue		$109,712
"Estimated Results Under Electrification:		
Gross earnings		$1,056,446
Operating expenses (66%)	$697,254	
Taxes	74,427	
		$771,681
Net revenue (electric operation)		$284,765
Net revenue (steam operation)		109,712
Increase		$175,053
Estimated cost of electrification		$8,000,000
Interest and depreciation 10%		$800,000
Saving in operation under electrification		175,003
Deficit		$624,947

55 "Our suburban traffic is not sufficiently dense to warrant the expense necessary to electrify these lines, and it is evident from the foregoing figures that even under electrification there would not be an increase in traffic sufficiently large to offset the annual loss from operation. It simply proves that under present conditions of cost of electrification of steam railways, where it means a replacement of a plant already installed, and serving the purpose, it is not justifiable to electrify either in whole or in part your Chicago terminals at this time."

56 The surburban district of the Illinois Central covers about 50 miles of road and carries in round numbers 15,000,000 surburban passengers per annum, or an average of 41,150 per day, or 1,700 per hour. An increase of 100 per cent in earnings would not enable the road to break even.

57 The Railway Age Gazette, in commenting editorially on Mr. Harahan's statement, says:

58 "It may be accepted as conclusively demonstrated that the New York Central and the New Haven roads are moving trains by electricity more economically than they moved them by steam in their suburban district. To enable this to be brought about, however, extremely heavy capital costs had to be assumed and the charges on these capital costs make the entire operating cost, including overhead charge, far higher than it used to be in the days of steam operation.

59 "For example, a standard express train of eight cars on the New Haven road pulls out of Grand Central station headed by two half-unit electric locomotives, each of which cost very nearly $40,000. The capital cost of the motive power of this train is in excess of $75,000 [the interest and depreciation amounting to $20 per day]—the brackets are ours. The cost of motive power at the head of a similar New York Central passenger train operated by electricity is about one-half this sum. Moreover, it will be recalled that Mr. Wilgus estimated that the direct costs of electrical equipment represented only one-fourth of the total charges attendant upon electricity. The cost of making everything ready and safe for this kind of operation is far greater than the highest estimates are apt to contemplate."

60 From a report of the Electrical Commission of the State of Massachusetts the following extracts are taken (letter of C. S. Mellen, president of the New Haven road):

61 "We believe we are warranted in saying that our electric installation is a success from the standpoint of handling the business in question efficiently and with reasonable satisfaction, and we believe we have arrived at the point where we can truthfully say that the interruptions to our service are no greater, nor more frequent, than was the case when steam was in use. But we are not prepared to state that there is any economy in the substitution of electrical traction for steam; on the contrary, we believe the expense is very much greater."

62 The Boston & Albany Railroad Company reports the result of their study and estimates the requirements as follows: A power station of 6000 kilowatts will be necessary, with storage batteries to handle the peak load. The total cost of the installation is estimated at $4,000,000, and the interest, taxes, and depreciation at 9 per cent, or about $400,000 per annum. A stock argument for electric operation is the saving to be made in operating expenses, but concerning this the following statement is made:

63 "Some slight economies would accrue in the transportation

expenses under this operation, which would be substantially absorbed by the additional expenses to be incurred for the maintenence of the additional apparatus installed, and the net economies would be so small as to be inappreciable in the consideration."

64 Another stock argument of the advocates of electric locomotives is the growth of traffic which is supposed to result from electric operation. This argument is met as follows in the report:

65 "Considering now the possibilities of increasing the traffic, the statistics of the B. & A. R. R. show substantially the following number of passengers handled in the above territory per annum:

1891	4,552,918	1899	3,897,364
1894	4,799,578	1907	4,435,841

66 "The absence of any material increase in traffic is probably due to the fact that the circuit is occupied as a high class residential district not susceptible of rapid subdivision of property, and more particularly to the fact that suburban lines are being rapidly extended into all such outlying districts and afford a more advantageous means of collecting and distributing local travel through the commercial and residential districts than could possibly be afforded by a railroad constructed and operated upon private right of way and devoted largely to long haul operations."

EXAMPLE TO ILLUSTRATE A CONCRETE CASE

67 The following illustration representing a concrete case is selected because of its elementary character, more especially as the case is so simple that all the variables affecting the comparison are eliminated and the amount of coal to perform the operation is directly known:

Conditions: trailing load 1,600 tons; average grade, 1.3 per cent.; distance, 8 miles; speed, 15 miles per hr. for electric and 14 miles per hr. for steam locomotive.

(*a*) Electric

1,600 net tons
190 Loco. (2) tons
———
1,790 gross tons

$$R = \begin{cases} 1.3\% \text{ grade} \times 20 = 26 \text{ lb.} \\ 5° \text{ curves} \quad 3 \text{ lb.} \\ \text{Level} \quad 6 \text{ lb.} \end{cases}$$

$$\frac{\text{Gross tons} \times R \times \text{Distance}}{500} = \text{kw-hr. at the rail}$$

Substituting values:

$$\frac{1{,}790 \times 35 \times 8}{500} = 1{,}000 \text{ kw-hr. (at rail)}$$

Equivalent kilowatt load at power house =

$$\frac{\text{Tons} \times R \times \text{m.p.h.}}{500 \times \text{Efficiency } \%}$$

Where the efficiency between the rail and generators equals 65 %, substituting as before:

$$\frac{1{,}790 \times 35 \times 15}{500 \times 65\%} = 2{,}900 \text{ kw.}$$

For this particular case current can be purchased from an adjacent power house at the very low rate of one cent per kw-hr. at the rail.

At this rate the power cost per trip will be 1,000 kw. at one cent = \$10.00.

(*b*) Under steam conditions we have the same as before, 1,600 net tons + weight of two locomotives, 300, or 1,900 gross tons.

The coal consumption for this particular run is 6,000 lb.

The price per ton to equal the electric cost for power, is:

$$\frac{6{,}000 \text{ lb.} \times \text{price per ton}}{2{,}000} = \$10.00$$

Transposing:

$$\frac{2{,}000 \times 10}{6{,}000} \times \$3.33$$

But as coal for this particular case costs the road \$1.70 per ton, the relative cost, coal against power, is

$$\frac{6{,}000 \times \$1.70}{2{,}000} = \$5.10$$

There is a difference in ton mile hours, in favor of the electric locomotive, due to speed and reduced gross tonnage, as follows:

$$\text{1st Electric} \quad \frac{1{,}790 \times 8 \times 8}{15} = 7{,}640 \text{ Gross ton mile hours}$$

$$\text{2d Steam} \quad \frac{1{,}900 \times 8 \times 8}{14} = 8{,}690 \text{ Gross ton mile hours}$$

To make the comparison correct the coal consumption of the steam locomotive should be proportioned on the ton mile hours, produced, and the cost of coal then becomes:

$$\frac{\$5.10 \times 8{,}690}{7{,}640} = \$5.80$$

Adding to the foregoing the other operating costs the relative expense becomes

(*a*) *Electric.* Power \$10.00

Lubrication, supplies, repairs, crew at \$0.1158 per 1,000 ton miles, or

$$\frac{0.1158 \times 1{,}790 \times 8}{1{,}000} = \dots\dots\dots \quad 1.66$$

Interest and depreciation, taxes, insurance, etc., at 10% .. 1.46

\$13.12

(*b*) *Steam.* Coal as above $5.80

Lubrication, supplies, water, repairs, enginemen at $0.25 per 1,000 ton miles,

$$\frac{\$0.25 \times 1,900 \times 8 \text{ miles}}{1,000} = \ldots\ldots\ldots\ldots \quad 3.80$$

Interest and depreciation at 10% (2 locomotives)

$$\frac{\$34,000 \times 10\% \times 8}{365 \times 24 \times 14} = \ldots\ldots\ldots\ldots \quad 0.22$$

$9.82

Cost per trip in favor of Steam, $3.30, or 25% less

EXAMPLES TO ILLUSTRATE A CONCRETE CASE

68 The idea is all too prevalent with the public, and even with some of the bodies that have been given legal power of supervision over railway companies, that any expenditure which can be forced upon the railway companies is just so much gain for the public. Never was there a more absolute fallacy. In the long run, the cost of every bit of railway improvement must be paid for by those who buy tickets and ship freight. Economy in the administration of our railways is just as important in the interest of the general public as if the railways were actually under government ownership.

Recently *The Engineer* (London) editorially made a plea for a "common denominator" for comparison of engineering achievements, using the following illustrations:

"Thus for example, if we take Mr. Humphrey's reply to Mr. Davey's criticisms, we see that he gained a mere dialectical advantage by showing on the screen a great differential pump, and beside it an internal combustion pump, so small by comparison that he had to explain that it was not a "hooter." Both engines could deal with the same quantity of water; but the Davey engine was lifting it 1500 ft. from a mine, while the gas pump could not lift it more than about 15 ft. Indeed, it could not do the work of the Davey engine at all."

Also a comparison was drawn between the cost of working with producer gas engines and steam engines. The argument was all in favor of the gas engine, expressed in weight of fuel required per hour to develop a horse-power. But the aspect of the matter changed when it was pointed out that the coal used by the steam engine was slack, costing $1.75 per ton, while the gas producer worked with anthracite, costing over $6.25 a ton. Here the cost of fuel was the common denominator, not the weight of the fuel.

The plea concluded by saying that the common denominator should be the commercial cost. E. H. McHenry expressed the same idea when he said that "Engineering is making a dollar earn the most interest."

LUBRICATION AND LUBRICANTS

By Charles F. Mabery[1]

Non-Member

Next to the conservation of the world's fuel supply there is probably no subject of greater importance in the manufacturing world than the control of waste power caused by imperfect lubrication and needless friction. Notwithstanding the increasing interest in more economical methods, the immense losses from this source are scarcely appreciated. In his recent work on lubrication and lubricants, Archbutt stated that with considerably more than half the 10,000,000 h.p. in use in the United Kingdom of Great Britain, 40 to 80 per cent of the fuel is spent in overcoming friction, and that a considerable proportion of this power is wasted by imperfect or faulty lubrication. On account of the great abundance of cheap fuel in the United States doubtless the conditions here are even less desirable. It is safe to state that losses from this source in this country are from 10 to 50 per cent of the power employed. Not infrequently in factories where the annual expense for lubrication amounts to thousands of dollars, lubrication experts find a loss of 50 per cent, or greater.

2 The manufacturer often knows very little concerning the economic qualities of the lubricants he receives; in using them, too much is left to "rule-of-thumb" methods with little knowledge of the actual conditions of friction, the action of metallic surfaces under the dynamic stress of the transference of power, or such modified action as is produced by the intervention of a lubricating film. For example, the different effects on a journal of a soft and hard bearing may be sufficient to cause a considerable loss of power if improperly selected, and yet may escape attention. In the earlier tentative study of the conditions depended on for the results described in this paper, under such loads as 100, or 150 lb. per sq. in. of bearing surface, the grades of babbitt in ordinary use were found much too soft and yielding to sus-

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tain such work under the necessary conditions of speed and oil feed; only a very hard alloy of exceptional composition could be used. The one selected of approximately the composition, Tin, 90; Copper, 2; Antimony, 8, gave results entirely satisfactory. Then since it was desired to maintain such conditions of load and speed that any oil could be broken down at any moment, it was found necessary, not only that the journal and bearing be milled to mechanically true surfaces, but that by continued operation and repeated careful milling, even a higher degree of permanent evenness be maintained. If such be the essential conditions in precise quantitative observations, similar precautions are evidently necessary in factory operations.

3 In the earlier days of machinery lubrication before the introduction into the trade of products from petroleum, the manufacturer had little concern about viscosity and other physical constants of lubricants, for, dealing with simple oils or greases of definite composition, he could be sure of obtaining what he desired within the capacity of the materials at his disposal. Then, in the days of higher prices of manufactured products and less severe competition, imperfect lubrication was of less consequence than in more recent times when every detail of cost and loss should properly receive careful attention; and, furthermore, the principles of friction and the importance of its control were only imperfectly understood in the earlier days of lubrication. Modern high speeds and excessively heavy loads had not then to be provided for in the applications of power in manufacturing operations, or in transmission or transportation.

4 The discovery that the heavy hydrocarbons in petroleum possessed the qualities requisite for lubrication—viscosity, durability and stability under varying conditions of speed and load—was the beginning of a new era in lubrication. Methods of treatment and refining, with little or no knowledge of the hydrocarbons of which the lubricating oils were composed, and developed entirely along empirical lines, were slow in producing suitable products. The earlier methods have undergone no fundamental changes even to the present time, except in the introduction of heavier hydrocarbons from crude oil territory more recently developed. Crude oils of the Pennsylvania type containing a considerable proportion of the hydrocarbons $C_n H_{2n+2}$ have always yielded excellent light spindle oils composed for the most part of the hydrocarbons, $C_n H_{2n}$, and $C_n H_{2n-2}$. But, as we now know, this type of oils include too small a proportion of the heavier hydrocarbons for the body necessary in lubricants subjected to the great stress of heavy loads and cylinder friction. This need in heavy

lubrication led to the practice of compounding oils, or mixing with the petroleum products various proportions of the vegetable oils, such as castor or rape, and the various animal oils or greases, which so fully monopolized this field, that manufacturers were often led to believe that no other products could serve an equivalent purpose. Even since the more recent introduction of heavy lubricants from Texas and California petroleum the belief still prevails that only compounded oils can be relied on for heavy work. But with care in distillation and treatment, it is certain that heavy lubricants, well adapted for bearings and cylinders, may be prepared from those crude oils, and large quantities of such lubricants are now widely in use.

5 All experimenters with lubricating oils who have given thoughtful attention to the essential needs of lubrication have been impressed by the superiority of an ideal solid lubricant, i. e., one that should embody an equivalent of the desirable qualities of the liquid products with a greatly superior wearing quality, a low coefficient of friction, and readily convertible into a form that can conveniently be applied to the various forms of journals and bearings. Soapstone, asbestos, natural graphite, etc., do not, altogether, possess these fundamental qualities of the liquid products. Greases compounded with graphite are useful on low-speed bearings and under heavy work. Natural graphite serves an excellent purpose on cast-iron bearings, acting as a surface evener of the porous metal. On finer surfaces care is necessary that it does not collect in such quantities as seriously to scratch or abraid the journal and bearing.

6 Of all the solid bodies available for lubrication, graphite possesses the desirable unctuous quality and great durability. For general use in lubrication, graphite must be in its purest condition and in a state of extreme subdivision. Whether, in such a condition as the deflocculated form, the ultimate molecules or atoms have a certain freedom of movement, analogous to that of liquid molecules under stress of friction, or whatever explanation may be suggested of its unctuous quality, the fact remains that it possesses this quality in very high degree. Such graphite is now produced by processes discovered, perfected, and placed on a manufacturing basis by Dr. Edward G. Acheson of Niagara Falls as a part of his great work in the development of electrochemical processes. Besides his immense output of pure graphite for general commercial use, Dr. Acheson has succeeded in converting it into a new form, a deflocculated condition, that meets the requirements of an ideal solid lubricant. This deflocculated form greatly surpasses ordinary graphite in unctuous quality,

and its adaptability for prolonged suspension in water and oils renders it especially applicable to frictional conditions. Furthermore, the readiness with which it forms coherent films on journals, its great wearing qualities and the ease of the application, constitute a lubricant of extremely high efficiency.

7 Acheson graphite can be produced from any substance that contains carbon in a non-volatile form. Under the extreme temperatures of the electric furnace any and all other elements are readily volatilized. Even carbon itself is freely vaporized and its peculiar appearance in the burning carbon-monoxid is depended on as an indicator of suitable conditions in furnace operation, much as the drop in the manganese flame which shows the disappearance of carbon in the Bessemer converter.

8 As commercial products two forms of graphite are produced, the unctuous and the deflocculated modifications, the first form accompanying the production of carborundum in furnaces charged with carbon and sand, the second obtained from a charge of coal or coke alone. The first form is leafy in structure, coherent, and extremely unctuous or greasy to the touch; it is segregated and not readily disintegrated. The second form is also unctuous in a high degree, but very pulverulent and capable of extreme subdivision; it is readily converted into a deflocculated condition. This form in water forms the commercial "Aquedag," or aqueous Acheson deflocculated graphite. In combination with oils it is known as "Oildag."

9 This deflocculated graphite has peculiar properties; it remains suspended indefinitely in water, but is quickly precipitated by impurities. On account of its extreme subdivision a very small amount suspended in water serves for efficient lubrication. From numerous and long-continued trials it appears that 0.35 per cent serves an adequate purpose and that a larger proportion is superfluous. It is certainly remarkable that such a small quantity of graphite is readily distributed by water between a journal and bearing while sustaining a load of 70 lb. per sq. in. of bearing surface, and that under high-speed conditions it maintains an extremely low coefficient of friction.

10 Proper lubrication of bearing surfaces involves careful consideration of the metals composing the journal and bearing, since the influence of the metals employed has an effect even in the intervention of the best lubricating film. The materials in common use for the construction of bearings include cast iron, steel, and alloys of variable composition included under the general terms, bronze and babbitt. In high-speed work cast-iron bearings must be used with extreme care.

In the accurate adjustment necessary in machine testing of lubricants, we have found it impossible to prevent injury to the journal when using a cast-iron bearing. Results obtained by the use of bronze have not been altogether satisfactory. However, properly selected babbitt for a steel journal seems to fulfill the desired conditions most satisfactorily and it possesses a wide range of applicability. As mentioned above, satisfactory lubrication is possible only when the journal and bearing are properly machined to true surfaces, kept smooth, accidental scratches worked out, and bare spots avoided. Successful lubrication demands constant skilled attention to the condition of journals and bearings, and no factory supervision affords more desirable returns. Lubrication consists in reducing friction to the lowest increment of the power in use. A lubricant is an unctuous body that readily forms a continuous, coherent, durable film capable of holding apart rolling or sliding surfaces, and itself interposing the least possible resistance. The economic problem in lubrication depends on the use of such a lubricant under suitable conditions.

11 The lubricants in commercial use include water, oils, greases and solids. Under oils are classified the great variety of light spindle, heavy engine and cylinder products, either unmixed hydrocarbons from petroleum or compounded oils, tallow, wool grease, etc. The greases may be generally classified under a few heads depending on their consistency, which is derived from the proportion of lime or soda soaps or oleates mixed with the hydrocarbon oil as a carrier. The solid greases have already been referred to.

12 Water in itself possesses no oiliness whatever but under certain conditions in cylinders it is found to assist in imparting to the metallic surfaces an extremely smooth condition which serves materially to reduce the friction. A practical knowledge of hydrocarbon lubricants should include a knowledge of the source, that is, the crude oil from which the lubricant is prepared, since there is a wide difference in composition and properties of the oils from different oil fields. Methods of refining petroleum oils have very much to do with the quality of the products. In general terms, inferior products are obtained when the process of distillation is conducted in such a way as to produce decomposition; the best products are obtained only by careful distillation and careful treatment in refining, whereby the hydrocarbons in the refined products obtained have essentially the same composition as in the original crude oil.

13 An examination of various lubricants in the trade frequently reveals a condition of the oils indicating improper refining. For

example, it does not need the application of extremely delicate tests to show the presence of free alkali, of sodium sulphate, or of sodium salts of organic acids, any one or all of which may be injurious to metallic surfaces. One of the most exacting duties of the refiner is the treatment with caustic soda in such a manner as to remove all acid products and at the same time to avoid such an excess of caustic as will form an emulsion, which is one of the "terrors" in a refinery. An examination of a great variety of oils in the trade, such for instance as the spindle oils in use in automobile service, indicates that the best refined oils are those that contain a minute trace of alkali.

14 The ordinary methods of testing lubricating oils include determinations of the viscosity, the specific gravity, the flash and the fire temperatures. Another important property of these oils, termed oiliness or greasiness, is not so readily determined by analysis; in fact, there seems to be no accurate method for its determination; yet it is readily distinguishable and has much to do with the efficiency of all lubricating oils. Concerning the most efficient methods of testing lubricating oils various opinions are expressed by different authors. Redwood, in his work on petroleum and its products, asserted that the viscosity of an oil is the best guide to its lubricating value since it enables the consumer to select oils similar to those that have afforded him the best practical results. He alludes to the close relationship between viscosity and the laws of friction of liquids. In comparing the use of viscosity with observations on the behavior of lubricants on a frictional testing machine, he states that he was unable to obtain satisfactory results with any machine at his disposal. His conclusions in general were that in the present state of our knowledge the indications afforded by testing machines are wholly misleading, and this led him to attach especial importance to a good system of testing viscosity. He refers to the opinion of Thurston that any oil should be tested on a machine under the conditions of load and speed similar to those of the use for which the oil is intended.

15 Referring to the work of Ordway and Woodbury in 1884 with an apparatus constructed to apply pressures of 40 lb. per sq. in.; to those of Tower carried on under what he terms great pressures—100 to 600 lb. per sq. in.— in an oil-bath system of lubrication; and also referring to the opinions of others on these results, Redwood presents the view that the agreement between machines and actual practice is extremely slight, his final conclusion being that viscosity affords the most valuable tests of lubricating qualities at our disposal. Inasmuch as Redwood's opinion on machine testing is a result of his observations

during several months on the Ingram and Stafer machines, in which the speed is 1500 r.p.m., and that the friction is gaged by the number of revolutions necessary to carry the temperature to 300 deg. fahr.: it is not difficult to understand his conviction that in his experience testing machines do not afford results comparable with those of actual practice.

16 The value of viscosity as a distinguishing property of lubricating oil is recognized by all who have given attention to the subject, but all are not agreed as to the extent of its practical reliability. Archbutt suggests that the quality of oiliness or greasiness is nearly of as much importance as viscosity. Although, as mentioned above, there is no precise method whereby oiliness can be determined, it is not difficult to recognize it nor to distinguish the marked differences in this respect shown by different oils and greases. Archbutt calls attention to the fact that at very low speeds the friction of a cylindrical journal should be proportional to the viscosity of the oil; but at higher speeds, and consequently increased temperatures, the relation of friction to speed ceases; the viscosity is diminished with a corresponding change in the carrying power of the journal. While fully appreciating the value of the information to be obtained by chemical analysis, Archbutt insists that the oiliness of a lubricant is of especial importance under heavy loads and high speeds. He suggests that it is advantageous for an engineer to test oils for himself on a machine without depending altogether on analytical data of physical tests obtained from the expert. Hurst also mentions that a broader knowledge of the practical working of oils is necessary than can be obtained from chemical or physical tests alone. He maintains that the test of an oil from a journal under the practical conditions of its use show conclusively its adaptability to such use.

17 The principal points to be observed in mechanical tests are the effects of speed, load, temperature, and the frictional effects due to viscosity and oiliness. The measurements on which depend the quality of the oil include the frictional resistance, the temperatures, and the endurance of the oil film. Doubtless the numerous machines that have been constructed for testing oils have certain merits and advantages. In the wide range of work carried on in this field during the past year, a part of the results of which are presented in this paper, the machine devised by Professor Carpenter was used. In its sensitive adjustment, durable efficiency, and the wide range of possible tests, this machine in continuous use during this period on light and heavy oils, greases and graphite, has fulfilled all requirements. Since

the results to be presented are closely dependent upon the method employed a view of this machine is here introduced.

18 This machine has an accurate adjustment for recording the speed, and a long lever arm with a vernier attachment graduated to tenths of a pound for recording the friction. The load is applied by a powerful spring worked by a cam and lever, the limit of the machine being 6000 lb., total load. Careful calibration of the spring showed it to be properly adjusted.

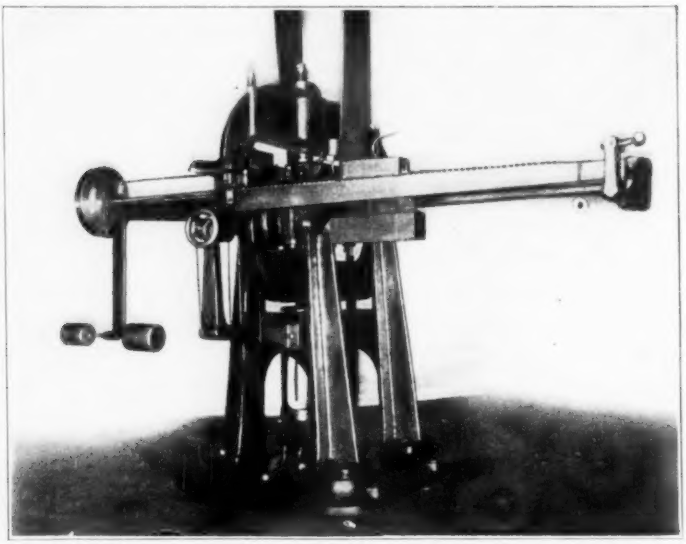


FIG 1 THE CARPENTER MACHINE FOR TESTING LUBRICANTS

19 In projected area the bearing in use is approximately 8 sq. in.; the journal is about 3 in. in circumference, nearly equal to 1 ft. in linear extension. A cast-iron frame babbitted and machined down to a true surface was used for the most part in this work. Even after careful machining some continuous frictional work was necessary on the babbitt surface to bring it to the proper conditions of constant results. The hard form of babbitt mentioned above gave satisfactory results, and there was little difficulty in keeping the surfaces in suitable condition after they were once obtained. For measuring temperatures a thermometer was inserted in a hole in the bearing, extending close to the journal.

20 Tests made at steam temperature—210 deg. fahr.—were carried on in a hollow cast-iron babbitted bearing, with steam attachments by which it was found that the desired temperature could readily be maintained. The lubricant is run in from a sight-feed cup through a small hole close to one side of the bearing with careful regulation of the flow for proper adjustment of the oil feed.

21 For delivery of the lubricant over the entire face of the bearing two channels or grooves are run diagonally across the babbitt face from the inlet hole, giving equal and even distribution; these channels must be carefully gaged for an even flow, otherwise dry spots or streaks appear on the journal accompanied by a sudden greatly increased friction indicated on the friction bar. This detail of operation requires careful and constant attention, for on it depends the continuous regularity of the friction curve. In this respect this method of observation is extremely sensitive, and is one of the important elements in frictional tests. Partial exposure of the journal enables the operator to observe the formation of the film, its comparative thickness and any irregularity due to an imperfect condition of the journal or bearing, or improper lubrication.

22 Accurate testing of the mechanical efficiency of oils with the precise quantitative observations possible on the Carpenter machine, including the various classes of lubricants under consideration in this paper, presented an extensive field of labor, especially since there are no general standards of comparison under any conditions of operation. Such constants must of necessity be based on arbitrary data, nevertheless if they are accurately determined on a standard machine, with the conditions of the journal and bearing selected,—the load and speed,—the constants on this machine may be readily ascertained on any other equally efficient machine. In duplicate tests made with the same bearing and under the same conditions the results were closely concordant. At the outset it should be clearly understood that these tests must be performed with a scientific accuracy of exact quantitative observations with close supervision of all details. The work then becomes the regular routine of any scientific investigation which involves long series of observations, after it is ascertained by preliminary trial what conditions are necessary in testing any given oil. Of course for commercial benefit these conditions should be as close as is practicable to the factory conditions of use.

23 The results to be described of the use of water, kerosene, and fuel oil as vehicles of graphite, present novel and interesting features. Under certain conditions, as mentioned above in steam cylinders, it is

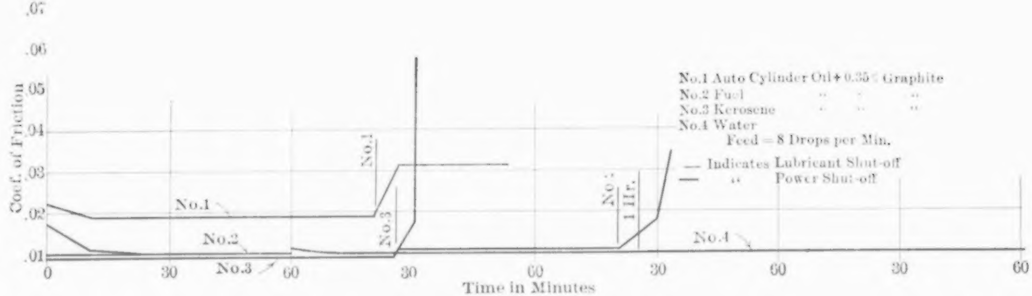


FIG. 2 CURVES OF FRICTION WITH VARYING VISCOSITY OF LUBRICANT. PRESSURE 70 LB. PER SQ. IN.; R. P. M. 446. No. 1 AUTOMOBILE CYLINDER OIL + 0.35 PER CENT GRAPHITE; No. 2 FUEL; No. 3 KEROSENE; No. 4 WATER. FEED = 8 DROPS PER MINUTE.

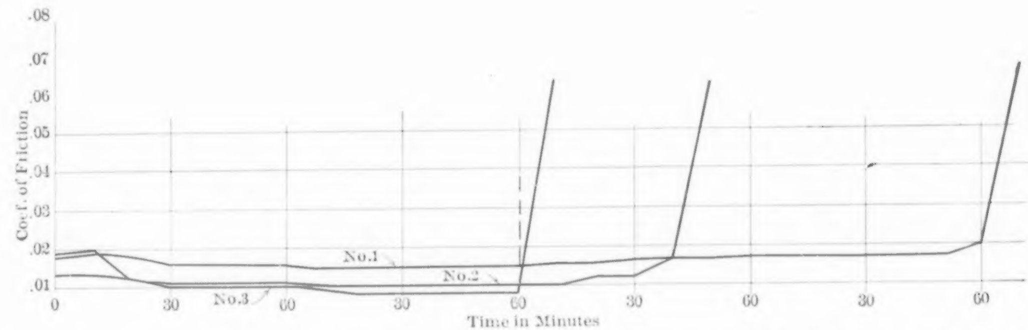


FIG. 3 CURVES OF FRICTION WITH VARYING VISCOSITY OF LUBRICANT. PRESSURE 150 LB. PER SQ. IN.; R. P. M. 445. No. 1 FAILS SPINDLE OIL + 0.35 PER CENT GRAPHITE; No. 2 FUEL; No. 3 KEROSENE. FEED = 8 DROPS PER MINUTE. DOTTED LINE INDICATES LUBRICANT SHUT OFF.

well known to engineers that water alone serves as a lubricating film. But since on journals it serves no purpose whatever, the lubricating qualities of aqueous suspended graphite must be due wholly to the graphite. The same is true of kerosene, which alone is practically devoid of lubricating quality, and likewise of fuel oils.

24 For the purpose of testing the effect of varying viscosity in lubricants, and at the same time the lubricating quality of deflocculated graphite, tests were made with water, kerosene oil, a fuel oil, and an automobile cylinder oil, each carrying 0.35 per cent graphite. The results obtained in these tests are shown by the curves in Fig. 2, in which the speed is maintained at 446, and the load at 70 lb. per sq. in. The observations of frictional load and temperature were made at intervals of ten minutes and on that basis a curve is drawn for each of the lubricants tested; on the chart the time is given in half-hour limits and the coefficient of friction in hundredths of a unit. It will be observed that the curve for water and graphite is practically a straight line, with scarcely any variation for the four hours shown on the curve; this test continued for 15 hours altogether with a precisely similar result. There were several stops which are indicated by a dotted line on the chart, and it appears that there was no change whatever in the direction of the curve by stopping and starting. Curve 3, representing the observations on the coefficient for kerosene oil with graphite, is also a straight line showing a coefficient very slightly lower than water. The coefficient curve for the fuel oil and graphite is also practically a straight line, and with an endurance test extending $1\frac{1}{2}$ hours after the oil supply was shut off; here the frictional coefficient is slightly higher than that of either water or kerosene. A similar regularity appears in the curve of the automobile cylinder oil with graphite; but it is to be noted that the frictional coefficient is very materially higher than those of the other lubricant shown on the chart, which may be considered as a measure of the comparatively greater internal viscosity of the automobile oil; this oil shows a much longer endurance test than appears on this chart.

25 The effect of varying viscosity in lubricants, and the lubricating quality of the graphite under practically the same speed, 445 r.p.m., but with a load of 150 lb. per sq. in.; using kerosene, a fuel oil and a spindle oil, with the same proportion of graphite, and the same oil supply, are shown on Fig. 3. Kerosene here shows a very slight irregularity in its coefficient, which differs only slightly from that on the preceding chart. Here again the greater internal viscosity of fuel oil is shown by the increased friction which appears in this curve. No

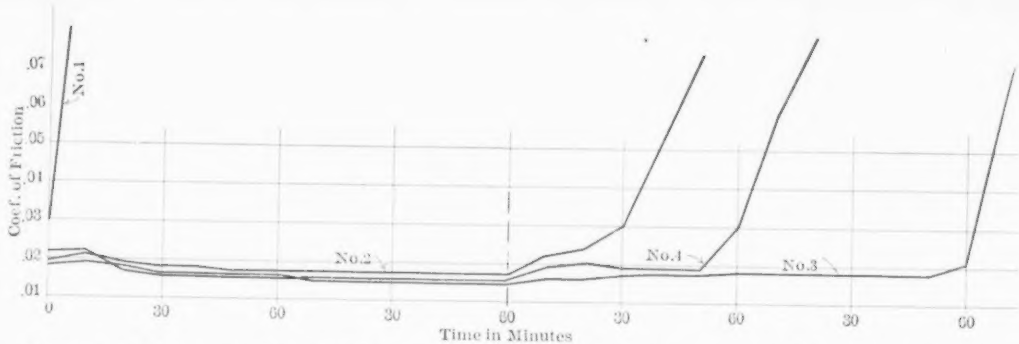


FIG. 4 CURVES OF FRICTION—OIL AND OILDAG—VARYING FEEDS. PRESSURE 150 LB. PER. SQ. IN.; R. P. M. 445. FAILS SPINDLE OIL.

No. 1	Oil alone	6 Drops per Minute	No. 3	0.35 per cent graphite	8 Drops per Minute
No. 2	" "	8 " " "	No. 4	" " "	4 " " "

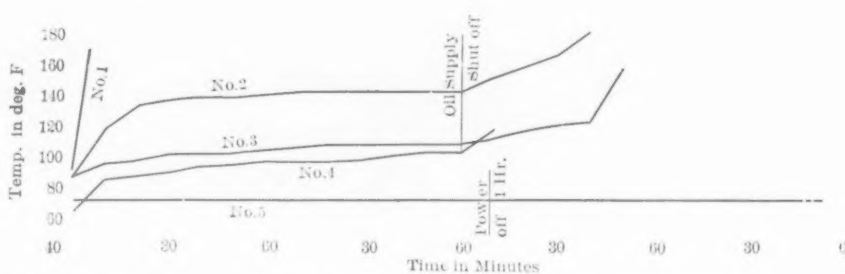


FIG. 5 TEMPERATURE CURVES FOR LUBRICANTS OF VARYING VISCOSITY, WITH AND WITHOUT GRAPHITE. PRESSURE 150 LB. PER SQ. IN.; R.P.M. 444

No. 1	Auto cylinder oil (alone)	6 Drops per Minute	No. 3	Fuel oil + 0.35 per graphite	8 Drops per Minute
No. 2	" " " + 0.35 per cent graphite	4 " " "	No. 4	Kerosene " " " "	8 " " "
			No. 5	Water " " " "	8 " " "

doubt the fuel oil possesses the quality of oiliness in a very slight degree, enabling it in the beginning of the test to take a lower coefficient than kerosene, which maintains a considerably higher coefficient for a few minutes, until the continuous film of graphite has formed and reduced the coefficient to its normal condition. It is evident that the fuel oil also possesses a certain oiliness which enables it to begin the test with a coefficient that changes only slightly during the entire period, including also an endurance test extending through two hours before the oil breaks and with only a slightly increased coefficient of friction after the oil supply was shut off. Another feature worthy of note is the comparative endurance of the three oils. While kerosene, under a bearing load of 150 lb. per sq. in., maintains an extremely low coefficient, the fact that it breaks immediately when the oil supply is shut off indicates that it has not the power to form a coherent graphite film, a power which is possessed to some extent by the fuel oil and in a marked degree by the spindle oil.

26 Fig. 4, load 150 lb. per sq. in., 445 r.p.m., gives the effect on a spindle oil of a variable feed. In one test on the oil alone the oil supply was regulated with the object of breaking the oil at the beginning of the test, and also its behavior was noted, under an oil supply that enabled it to perform its functions as a lubricant. The effect of graphite on the lubricating quality of the oil is also shown in Curve 3 and Curve 4, Curve 3 representing a feed of 8 drops per min., Curve 4 representing a feed of 4 drops per min. The diminished coefficient in Curve 4, as compared with Curve 2, represents the lubricating effect of graphite, and this effect is still further shown by the increased endurance test in Curve 4; it will also be observed that besides showing diminished friction, Curve 4 is based on an oil supply due to the graphite, one-half that of Curve 2 of the oil alone.

27 In Fig. 5 curves are shown which represent the temperatures recorded in tests of friction presented on Fig. 2 and Fig. 3. As in the previous charts the load is given as 150 lb. per sq. in. for the automobile oil, fuel oil and kerosene, and 70 lb. per sq. in. for water. The speed was 444 r.p.m. in all but the test with water, where it was 446 r.p.m. The test of the automobile oil alone showed an immediate rise in temperature, corresponding to the breaking point of the oil, which is shown in the friction test. It is interesting to compare this temperature with that of Curve 2, automobile oil and 0.35 per cent graphite, in which the temperature rises within twenty minutes to a definite point and then continues in a nearly straight line with little variation to the point where the oil supply was shut off at the end of two hours.

Curve 3, representing the temperatures of fuel oil and graphite, also shows a very slight variation after 30 min., when the stable conditions of lubrication were established. A difference in temperatures of approximately 25 deg. is shown between the curves of the automobile and fuel oils, which must represent the larger escape of energy in the form of heat from the bearing, due to the greater internal resistance of the automobile oil. The temperatures of kerosene with graphite, as shown in Curve 4, are approximately 10 deg. lower than those in the fuel oil curve, due to the still smaller internal resistance of kerosene. Bearing in mind the small difference between the specific gravity of the fuel oil, approximately 35 deg. Beaumé, and that of kerosene, approximately 45 deg. Beaumé, the difference in temperatures of these two curves is a good example of the accuracy in observation possible in these tests. Perhaps the most striking feature on this chart is the curve presenting the temperatures for water and graphite; here, as in the curve of friction for water, this curve is shown for only four hours, but the test actually extended through a period of 15 hours, during which time there were several stops in which, as shown on this chart, the temperature at the start was the same as that at the time of interruption. It will be observed that this chart shows an extremely low temperature, 65 deg., practically the same as the room temperature, which it never exceeded by more than 5 deg., and that it is essentially a straight line from start to finish. In this use of water as a vehicle for the graphite there is nothing to interfere with the best work that the graphite is capable of performing.

28 Among the various classes of lubricating oil examined in this work considerable attention has been given to the behavior of heavy engine and cylinder oils, both straight hydrocarbon oils and compounded oils. A special form of bearing was constructed, consisting of a cast-iron frame with a hollow chamber for introducing steam, and a babbitted face using the exceptionally hard babbitt previously described. In some of these tests a bronze bearing similarly constructed, but maintaining the bronze face, was employed. But in general it was observed that the results were less satisfactory with the bronze than with the babbitt bearings, in testing not only the heavy oils but the other classes of oils examined. Hard babbitt seems to possess certain peculiar qualities adapted to the various details and variations in speeds, loads, and temperatures, which are not found in the same degree in the bronze alloys. To show the results obtained in testing cylinder oils, charts are here presented on three commercial products, the American cylinder oil, Galena cylinder oil, and "600 W" cylinder

oil. Tests were also made on the influence of graphite on these oils, with reference to the frictional coefficient and endurance of the oils. The physical constants of the oils are also given for comparison, especially of specific gravity and viscosity. The general procedure of the tests included a continuous run for two hours at which time the supply of oil was shut off.

29 In Fig. 6 of the American cylinder oil, which is a straight hydrocarbon oil, the data of the tests include the use of the bronze bearing, a supply of lubricant at the rate of four drops per minute, a total pressure of 1200 lb., and a speed of 245 r.p.m. The curve of the straight oil begins at a somewhat higher coefficient that is maintained

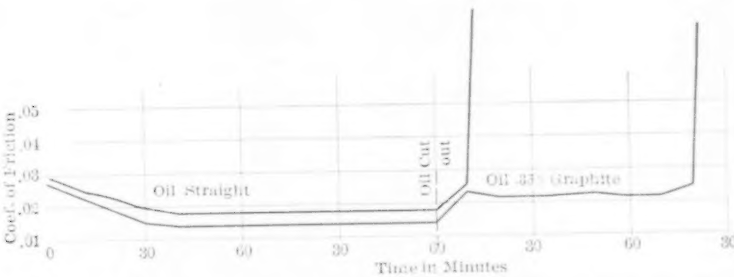


Fig. 6 American Cylinder Oil, with and without Graphite. Bzonze Bearing; 4 Drops per Minute; 1200 lb. Pressure; Temperature 210 deg Fahr.; Viscosity 100 deg. at 212 deg. Fahr.; Flash 440 deg. Fahr.; Specific Gravity 0.961.

after the first half-hour, when normal conditions are established, and it then proceeds in a straight line with no variation to the point where the feed is stopped. The endurance run of this oil is doubtless considerably shorter than it would have been with the use of babbitt bearings; in fact this was demonstrated in another test in which babbitt was used. With graphite the oil follows closely the direction of the other curve but with a very considerable diminution in the coefficient of friction. It further appears in the endurance test that the graphite carries the load with slightly increased friction for a period of 1 hr. 20 min., which would doubtless have been considerably prolonged if babbitt had been used.

30 Fig. 7 presents results obtained in tests of the "600 W" cylinder oil, with and without graphite. A comparison of physical constants with those in Fig. 6 shows a materially lower specific gravity and somewhat higher viscosity. In these tests the same total

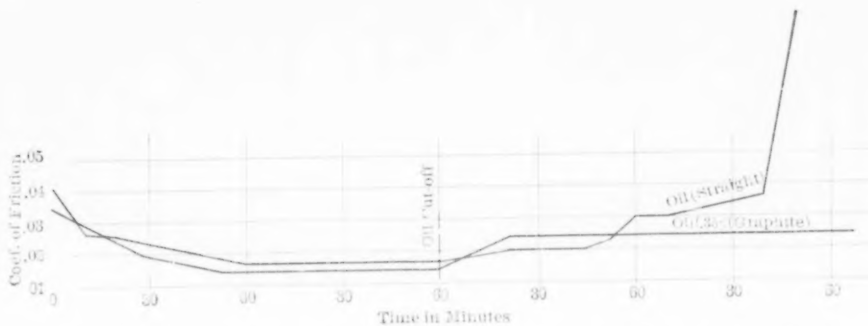


FIG. 7 "600 W" CYLINDER OIL, WITH AND WITHOUT GRAPHITE. BABBITT BEARING; 8 DROPS PER MINUTE; 1200 LB. PRESSURE; TEMPERATURE 210 DEG. FAHR.; VISCOSITY 150 AT 212 DEG. FAHR.; SPECIFIC GRAVITY 0.903; FLASH 530 DEG. FAHR.

FIG. 8 GALENA CYLINDER OIL WITH AND WITHOUT GRAPHITE. BABBITT BEARING; 8 DROPS PER MINUTE; 1200 LB. PRESSURE; TEMPERATURE 210 DEG. FAHR., VISCOSITY 116 AT 212 DEG. FAHR.; SPECIFIC GRAVITY 0.947; FLASH 266 C.

pressure, 1200 lb., and the same speed, 245 r.p.m., were used, but the oil feed was double that in the preceding tests and the babbitt bearing was employed. On account of the greater viscosity the straight oil showed at the beginning a considerably higher coefficient and the tests continued one hour before the oil had reached normal conditions, which it maintained until the feed was stopped and doubtless would have continued indefinitely. After the oil was shut off lubrication was maintained with some slight irregularity and increased friction during 1 hr. 40 min., the point at which it broke. Similar conditions are observed in the curve which expressed the variation in the coefficient of friction of this oil with 0.35 per cent graphite; it begins the test with a somewhat lower friction, reaching normal conditions sooner than the straight oil, continues in a straight line to the point where the supply is stopped, and then still continues in a straight line with somewhat increased friction. The endurance curve would doubtless have continued for a considerably longer time but the power was shut off at the point where the curve terminates. A marked influence of graphite on the behavior of this oil is plainly apparent in a comparison of these curves.

31 In applying tests to the Galena cylinder oil, with and without graphite, the same feed, load and pressure were used as with the preceding oil and the tests were made on a babbitt bearing. In viscosity this oil is somewhat less than the preceding oil, the specific gravity somewhat higher. Both curves in Fig. 8 begin with slightly lower coefficient at 0.03, and this difference is maintained until the oil is shut off and for $1\frac{1}{2}$ hours on the endurance test. To reach normal conditions the straight oil ran for 1 hr., the oil with graphite 45 min. After the feed was stopped, the curves proceed regularly with slightly increased friction, the oil alone practically breaking in $1\frac{1}{2}$ hours, the oil with graphite proceeding with perfect regularity for three hours, changing slightly during the next hour and breaking at the end of $4\frac{1}{2}$ hr. The tests represented in Figs. 6, 7 and 8 are not intended to present the comparative efficiency of these particular oils but to demonstrate the application of this method of testing and also to compare the effects of deflocculated graphite.

32 The results presented in this paper, with reference to the uses of graphite as a solid lubricant, indicate that in the deflocculated form it can readily be applied with great economic efficiency in all forms of mechanical work. One of its most characteristic effects is that of a surface-evener, by forming a veneer, equalizing the metallic depressions and projections on the surfaces of journal and bearing; and being

endowed with a certain freedom of motion under pressure, it affords the most perfect lubrication. In automobile lubrication the great efficiency of graphite, in increasing engine power, in controlling temperatures, and in decreasing wear and tear on bearings, has been brought out in a series of tests conducted by the Automobile Club of America. In connection with the reduction in friction of lubricating oils by graphite the extremely small proportion necessary is worthy of note; the proportion used in this work is equivalent to one cubic inch of graphite in three gallons of oil. The curve of temperature for Aquadag, an increase but slightly above that of the surrounding atmosphere, demonstrates an important economic quality of controlling temperatures in factory lubrication, thereby avoiding the danger of highly heated bearings, which are frequently the cause of fires.

33 In the observations described in this paper, and in fact in all the work that has been done in this field, there is not a more impressive example of the efficiency of graphite in lubrication than that presented in the curves of friction and temperature of water and graphite; for water serving merely as a vehicle and completely devoid of lubricating quality, the graphite is permitted to perform its work without aid and with no limiting conditions.

DISCUSSION

TAN BARK AS A BOILER FUEL

By David Moffat Myers, Published in The Journal for October

ABSTRACT OF PAPER

The average fuel value of spent hemlock tan is about 9500 B.t.u. per lb. of dry matter, which is about 35 per cent of its total moist weight in the fireroom. The available heat value per pound as fired is 2665 B.t.u. One ton of air-dry hemlock bark produces boiler fuel equal to 0.42 tons of 13,500 B.t.u. coal. The degree of leaching does not affect the number of heat units per pound of dry matter, but of course reduces the available material.

Boiler tests under normal conditions show thermal efficiencies of from 58 to 68 per cent, and a higher efficiency has been obtained under special conditions.

Tan presses have produced no marked increase in boiler and furnace efficiency when tested; with the same efficiency, however, an increase of about 5 per cent of steam for the same amount of tan bark may be expected owing to the increase of available heat units in the "tan as fired." Grate surfaces should be materially reduced when tan is pressed.

Mixing coal with tan under proper conditions increases both the capacity and the efficiency of boiler and furnace.

Conditions productive of best results have been: ample combustion space, and a refractory arch over the entire grate; no less than 0.5 in. and preferably of 0.6 in. water-gage draft with ample draft passages; feeding through holes in top of furnace in small quantities and at frequent intervals to approximate the rate of combustion; constant care to prevent blow-holes; a small shallow-fired furnace; a high arch above the fire (which is about the most important single requirement); proper ratio of heating surface to grate surface for local conditions; the pressing of tan under certain conditions.

ADDITION TO PAPER BY AUTHOR

The following illustrations and descriptive matter have been added by the author and should therefore be considered a part of the paper as presented at the December meeting.—Editor.

73 The following illustrations, reproduced from working drawings and sketches, will give some idea of the construction of furnaces for burning spent tan bark, sawdust, and bagasse.

74 Fig. 2 is a working drawing of the self-feeding tan-burning furnace designed by the writer and referred to in Par. 12 and Table 4.

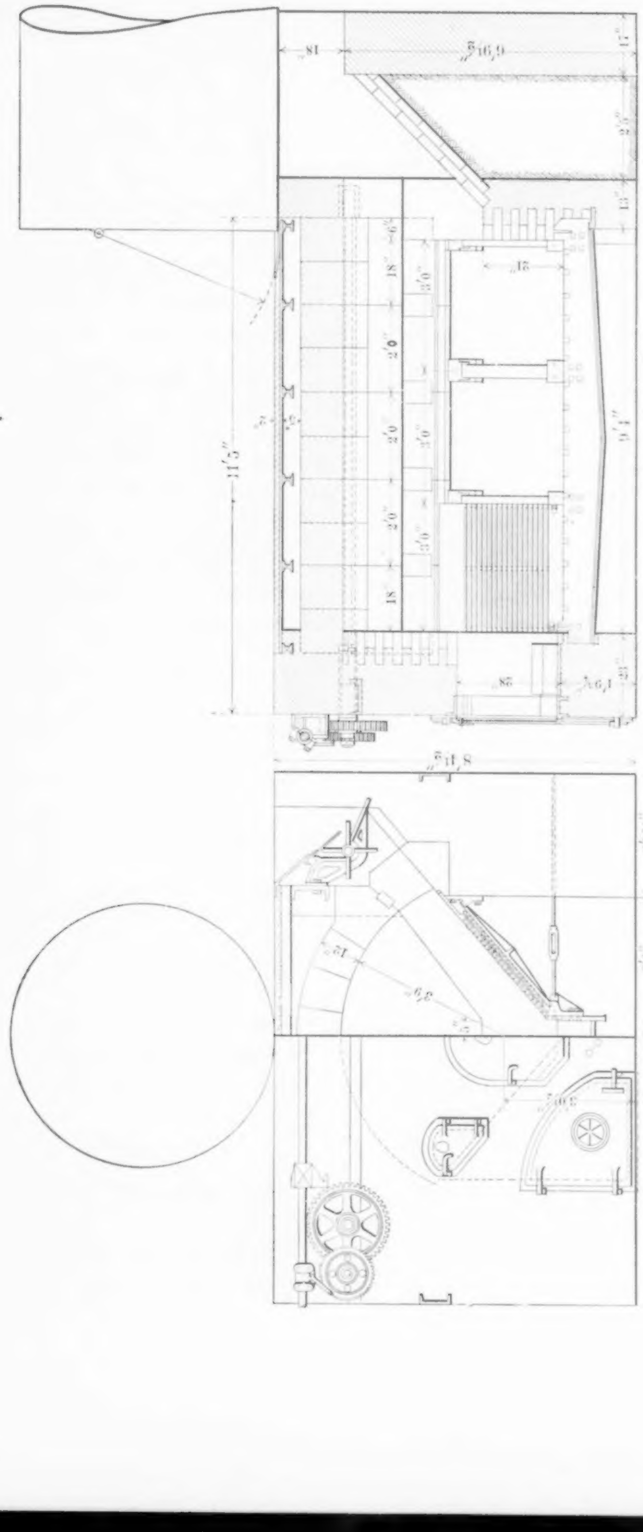


Fig. 2 End and Side Sectional Elevation of the Myers Tan-Bark Furnace

75 Fig. 3 shows the construction of the grate bars, which provide the horizontal draft opening tending to produce the draft action referred to in Par. 13.

76 Fig. 4 shows the application of these grates to bagasse burning. The stokers are done away with in this case, the fuel being fed by gravity to the feed chutes with weighted flaps which are used all over the islands of Cuba and Porto Rico. This burner has not yet been applied to bagasse burning.

77 Fig. 5 and Fig. 6 show the types of tan furnaces found by the writer in common use throughout the country. Fig. 5 shows what was known as the old Hoyt furnace. It was originally designed when tan bark was so plentiful that it was necessary to burn it. The writer has found these furnaces with inside lengths as great as 24 ft. on the grate surface. Fig. 6 shows a more modern type of burner designed to give a more even distribution of the fuel on the grates.

Fig. 3 Detail of the Grate Bars of the Myers Furnace

78 Fig. 7 shows a more up-to-date furnace designed for the hand firing of a mixture of coal and tan, the coal being mixed with the tan before entering the furnace, which is supplied with shaking or shaking and dumping grates. When coal is mixed with tan in any considerable proportion, more air is required for combustion, the best air spacing in the grate bar being found to be $\frac{3}{8}$ in. The percentage of draft area for this purpose should be about 40 to 50 per cent, depending upon how large a percentage of coal is used with the tan.

79 Fig. 8 shows what is known as a hump-back grate, which has been installed in different tanneries for the purpose of increasing the consumption of fuel in a given furnace. For instance, in a plant that had trouble in consuming all its tan bark, the writer merely took out the grate bars and put in a ridge bar as shown and converted the grate surface into the hump-back form. The result was that the consumption of tan bark per furnace was increased from 12 tons per day on the dry bark basis to 15 tons.

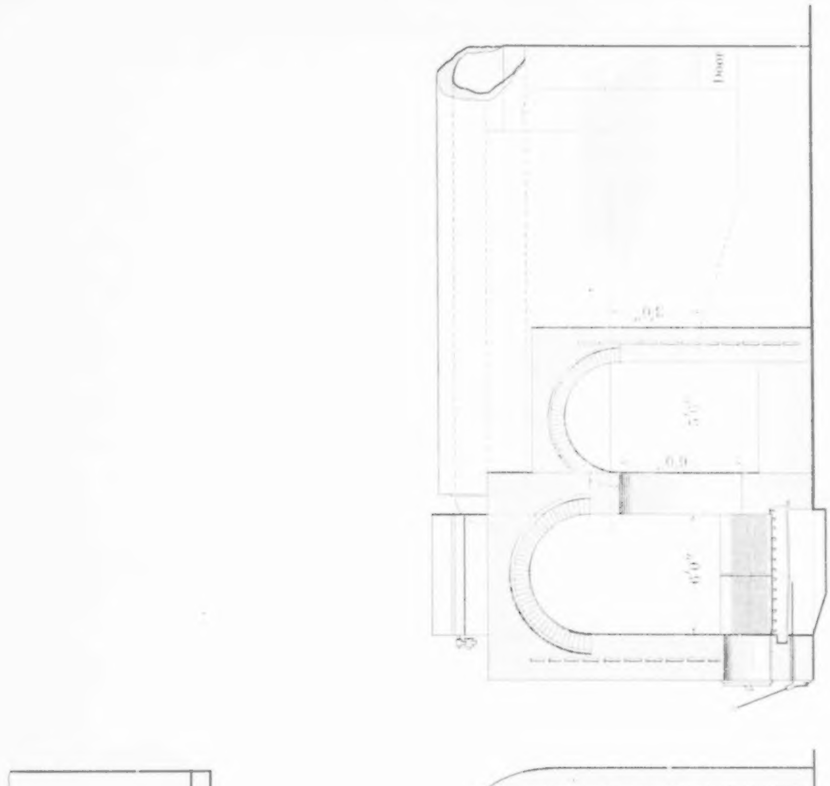


FIG. 4 PARTIAL PLAN, SIDE AND END ELEVATIONS OF THE MYERS FURNACE FOR BURNING BAGASSE

80 Fig. 12 shows what was known as the Thompson type of tan furnace. The MacMurray furnace, with a convex grate surface and feed pipes, is a type quite a number of which the writer has seen in operation.

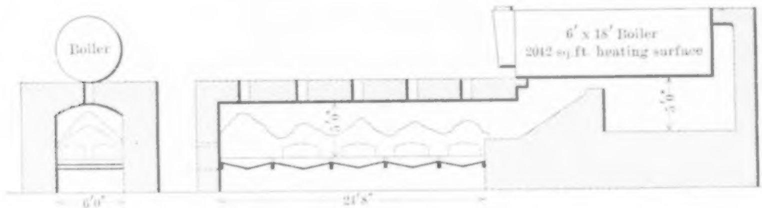


FIG. 5 THE EARLY HOYT FURNACE FOR BURNING TAN BARK

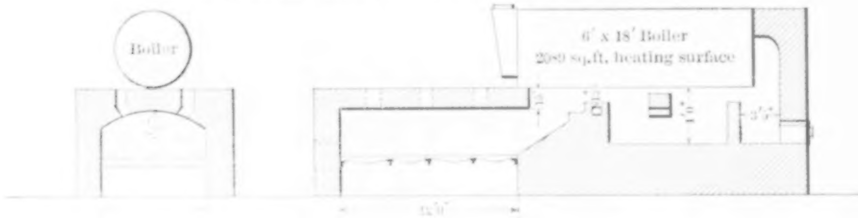


FIG. 6 A TAN FURNACE WITH SIX FEED HOLES

THE SETTING HAD AIR ADMISSION IN THE BRIDGE WALL AND A BAFFLE ARCH IN THE COMBUSTION CHAMBER. VERY GOOD RESULTS WERE OBTAINED.

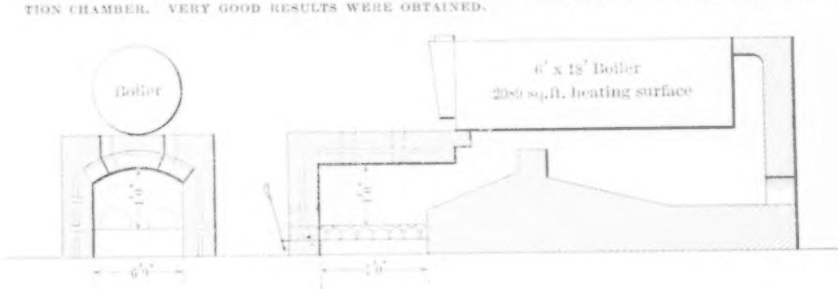


FIG. 7 A FURNACE WITH SHAKING GRATES FOR BURNING A COAL AND TAN MIXTURE

AIR SPACES OVER FIRE ARCH AND IN WALLS OF FURNACE AND BOILER WALLS. DISTANCE FROM GRATE TO TOP OF ARCH INSIDE SHOULD NOT BE LESS THAN 4 FT.

81 Fig. 10 is another form of tan furnace which gave good results in a plant in the South. The hump-back form of grate is reversed something like that used in the writer's stoker furnace, except that the tan is fed through a number of feed holes along the upper edges of these grates. This furnace was designed by the foreman in a Southern tannery.

82 Fig. 11 shows a design of the writer's for an adjustable gravity-feed furnace for burning tan or sawdust. The feed chutes are rectangular in section and contain adjustable chutes to regulate the depth of tan on the grates for any condition of draft, etc.

DISCUSSION

Albert A. Cary. The furnace described by Mr. Myers consists of an extension in front of the regular boiler setting, with a number of circular stoke holes, or openings through the top arch, over the

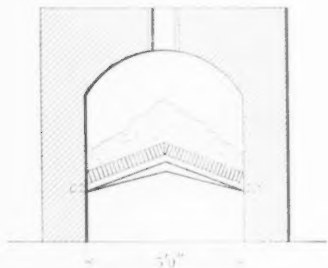


Fig. 8 Cross Section of Furnace with Hump-Back Grates and Bearing Bar

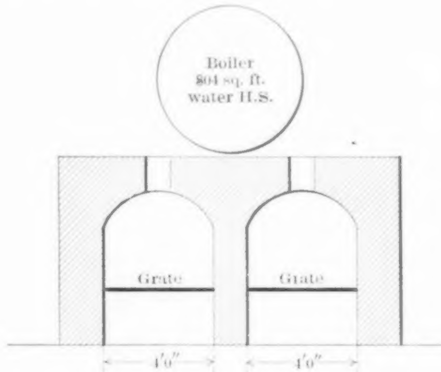


Fig. 9 Cross Section of a Double-Arch Tan Furnace of the Thompson Type on which a Test was Run

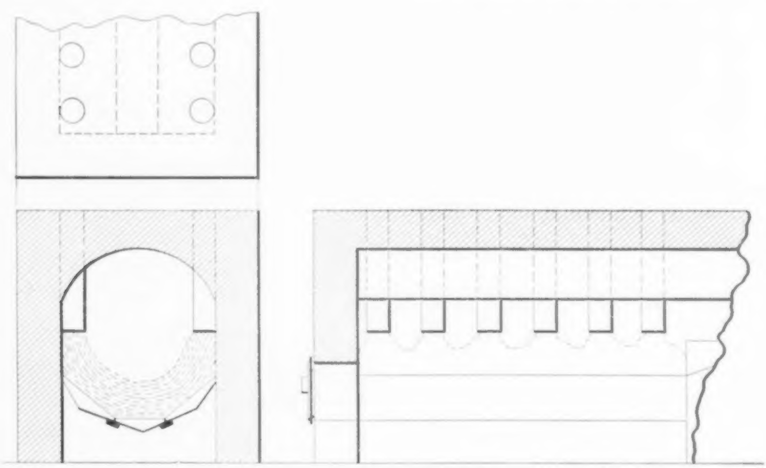


Fig. 10 Bush Tan Furnace with Multi-Tube Feed.

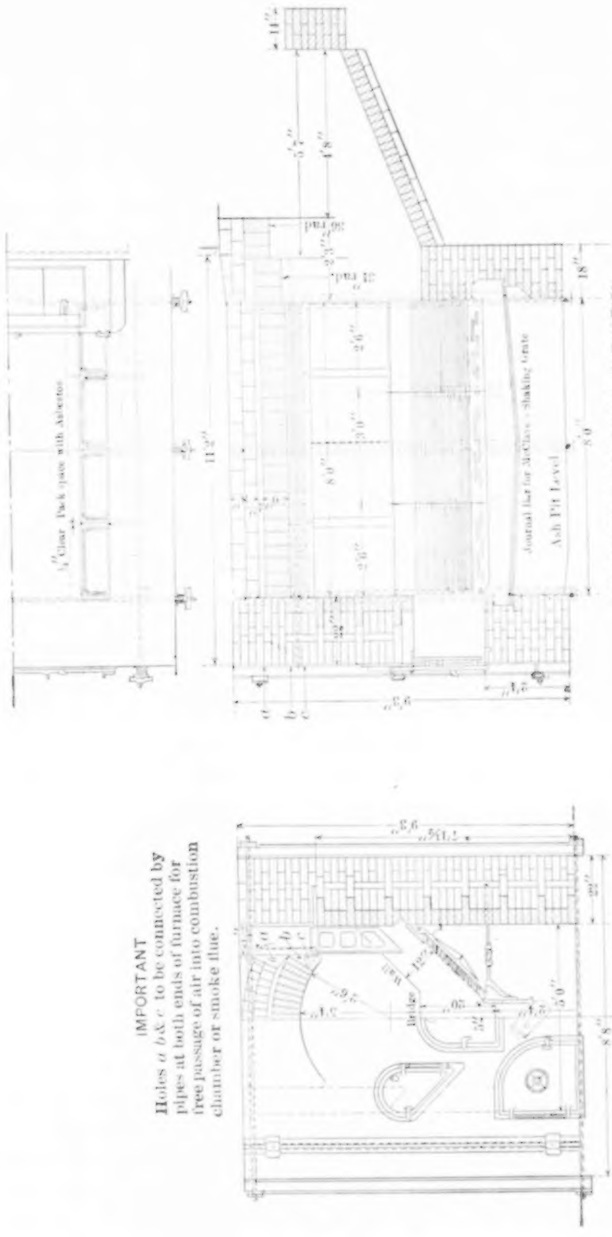


FIG. 11 A MYERS FURNACE WITH ADJUSTABLE GRAVITY FEED. ADAPTED FOR BURNING TAN BARK OR SAWDUST

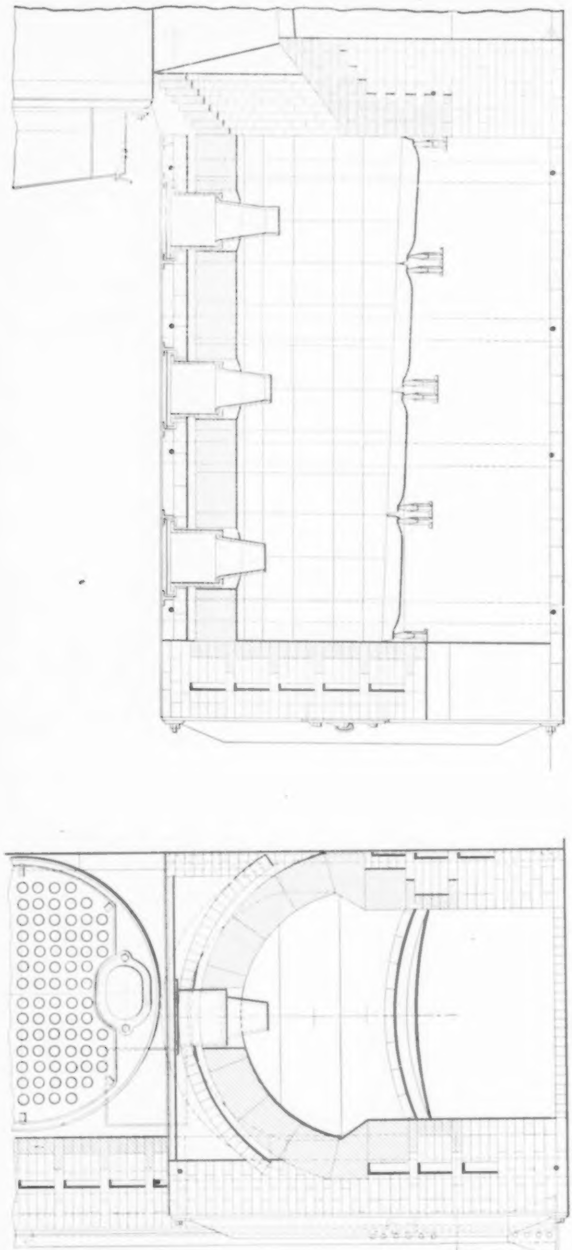


Fig. 1 The McMurray Furnace for Burning Spent Tan Bark.

grate. No little trouble has been experienced with this construction, due to the destruction of the lower end of these circular fire-brick tubes through which the fuel is charged to the furnace.

2 If these stoke holes could be always completely filled with fuel, so as to prevent inrushes of air, this destructive effect could be materially checked. However, as the method of charging fuel by hand is an intermittent one, the upper end of the cone of spent tan bark, soon after charging, drops below the level of the top of the arch, the inrushing air meets the hot furnace gases at these points and intense combustion results. For this reason, and due to the fact that when the excessive moisture in the fuel rises as a vapor against the arch, rapidly abstracting its heat (to become superheated steam), the fire brick cracks and disintegrates, finally resulting in a chipping off of the brick-work of the reverberatory arch around the lower end of the stoke holes. Repairs are therefore frequently necessary.

3 A continuous automatic feeding device, which would keep these stoke-holes constantly filled with the moist fuel, would undoubtedly do much to relieve this trouble by preventing an excessive infiltration of air at frequent intervals of time. Mr. K. McMurray of New York, has devised a very ingenous method for overcoming this trouble in hand-stoked furnaces. Fig. 1 shows both front and side sectional elevations of this furnace.

4 In the stoke hole is fitted a circular lining of cast iron which does not extend to the level of the inside of the arch. The lining is finished with a shoulder which diminishes the diameter of the opening by about two inches. A tube or open thimble drops into this frame, being held by a rim cast around its upper end. The lower end of the thimble extends about a foot into the furnace.

5 The fuel charged into the stoke hole falls through the thimble, and forms a cone-shaped pile below it on the grates. When the stoke hole becomes uncovered, the in-rushing air causes the intense combustion to take place, not on a level with the brick-work, but at a level below the thimble, and the life of the fire-brick arch at the stoke-hole openings is thus greatly prolonged. The ends of the cast-iron thimbles burn off gradually, but they cost but little, and are easily pulled out and new ones are inserted in their place.

6 Another trouble met with in this type of furnace is the rapid burning away of the fuel next to the side walls and the consequent large infiltration of air from the ash pit. This trouble has been largely overcome by reducing the width of the furnace by about 1 foot at the grate level, as shown in the front sectional elevation. The ledges

formed on either side of the lower part of the furnace support the cone of charged fuel on each side, thus keeping the grate effectually covered with fuel.

7 In this construction it will also be seen that instead of using a flat grate, the grate bars are curved so that the grate surface is higher at the center of the furnace than at the sides. This design decreases the thickness of the fuel bed under the stoke holes and causes a thickening of the fuel bed at the sides of the furnace.

8 Since water can be evaporated in the furnace itself only at a great loss, every practicable facility should be utilized for depriving the wet fuel of its moisture. Mr. Myers has mentioned the comparatively small gain from pressing the moisture out of the spent tan. I have used special rolls for extracting the water from moist fuels, with a desirable gain resulting. These rolls are of cast iron and run in pairs, one roll being about 12 in. in diameter, the other about 14 in. and both held together by heavy springs. As both rolls are revolved at the same number of revolutions per minute their surface speeds are necessarily different. The faces of the rolls are roughened by having a shallow checker work pattern cast upon them. The fuel is fed to the rolls continuously, and due to the tearing or macerating action between the rolls faces more than double the amount of water is thus worked out, as compared with the press results given in Mr. Myers' paper.

9 In one case, where the chimney was located some distance from the boilers, a wrought-iron rectangular flue was used to connect them, a shallow iron trough being formed on the surface of the flue by having the edges on the two vertical sides continued above the level of the top. The other three sides of the flue were covered in the usual way. The moist fuel was fed upon the chimney end of the flue and was drawn by a conveyor towards the boiler and over its top, whence it was delivered on top of the extension furnace. A small evaporation of moisture took place, sufficient to make this device desirable. The heat from the top of the boiler and the extension furnace may also be used in this way. The waste heat from the boiler may also be used to pre-heat the air delivered to the ash pit. I know of no condition where pre-heated air can be used to better advantage than with moist fuels.

10 Mr. Myers has spoken of the advantage of the high furnace over the low furnace. My experience thoroughly endorses this. When the moist fuel is charged into the hot furnace, a cloud of steam is evolved, which, when crowded down upon the burning fuel in

a low furnace, hinders combustion. A sufficient amount of steam would eventually extinguish the fire.

11 In addition to the effect of moisture described in Par. 7 and Par. 8, the large space occupied by the steam in the combustion chamber interferes with the combination of oxygen and the combustible gases evolved from the fuel.

12 In the flue-gas analysis obtained with moist fuels, of course the water in the gases condenses and is not accounted for in the analysis given.

William Kent. I consider this the most important paper on the subject of tan bark as a boiler fuel which has appeared in over thirty years. The only other paper that I know of is one by Prof. Thurston published in 1874 in the Journal of the Franklin Institute. He made some boiler tests on tan bark for fuel, using two different styles of furnace, some of his results being better than those given by Mr. Myers. I think that still better results are yet to be obtained from the use of tan bark as a fuel, by compressing out as much as possible of the moisture and using the waste heat of gases to dry the bark before it is put in the furnace. For burning the bark we must have a large fire-brick combustion chamber and give plenty of time to the burning of the gases, and then we will get as near the theoretically possible economy as can be expected.

2 The principal cause of poor economy in the burning of tan bark, besides the difficulty of securing good combustion in the furnace, is the amount of heat that is carried away in the shape of superheated steam in the chimney gases. If the bark, after being partly dried by compression, were further dried in a rotary drier by the waste heat from the chimney gases, there would be a very important gain in economy.

3 I have made a calculation showing the theoretical results that may be obtained in burning tan bark of different degrees of moisture under certain assumed conditions, the results of which are given herewith: The dry bark is assumed to have the following composition C = 0.50; H = 0.06; O = 0.40; N and ash = 0.04. Substituting in Dulong's formula, $14{,}600\ C + 62{,}000\left(H - \frac{O}{8}\right)$, its heating value is 7920 B.t.u per lb. Bark containing 20 per cent moisture would have a heating value of $0.80 \times 7920 = 6336$ B.t.u.

4 Assuming the chimney gases to escape at 600 deg., the heat required to evaporate the water from 62 deg. and to superheat the

steam to 600 would be $(212-62) + 970 + 0.48\ (600-212) = 1306$, or for 20 per cent moisture, 261 B.t.u. per pound of tan.

5 The 0.06 lb. of H in a pound of dry tan will unite with $0.06 \times 8 = 0.48$ O, making 0.54 lb. H_2O, which escapes as superheated steam carrying away $0.54 \times 1306 = 705$ B.t.u. for each pound of dry tan or $0.80 \times 705 = 564$ B.tu. for tan with 20 per cent moisture

6 Assuming 25 lb. of air to be required per pound of C + H in the fuel or $25 \times 0.56 = 14$ lb. of dry tan, the heat carried away by this air heated to 600 deg. is $0.24 \times 14 \times (600-62) = 1808$ B.t.u. per pound of dry tan or 1446 B.t u. for tan with 20 per cent moisture. Using the figures thus found the following table is constructed:

Moisture	B.t.u. per lb. wet tan	Losses of heat due to			Sum of losses	Net heat value B.t.u.	Efficiency per cent	Lb. Evap. per lb. wet tan
		Moisture	H in fuel	Heating air				
0.20	6336	261	564	1446	2271	4065	64.2	4.19
0.30	5544	392	493	1266	2151	3393	61.2	3.50
0.40	4752	522	423	1085	2030	2772	57.3	2.81
0.50	3960	653	352	904	1909	2051	51.8	2.11
0.60	3168	784	282	723	1789	1379	43.5	1.42
0.70	2376	914	211	542	1667	709	29.8	0.73
0.80	1584	1045	141	362	1548	36	2.5	0.03

7 Suppose that tan with 60 per cent moisture were dried to 20 per cent before being put into the furnace, using for this purpose the waste heat of the chimney gases, we would then have 0.40 dry tan + 0.60 moisture dried to 0.40 dry tan + 0.10 moisture, 0.50 water being removed. Suppose the moisture and the waste gases left the drying chamber at 300 degrees. Each pound of water dried out would take $(212 - 62) + 970 + 0.48\ (300 - 212) = 1162$ B.t.u. and 0.5 lb. would take 581 B.t.u. The H in the 0.40 lb. of dry tan would make 0.216 H_2O, which would take away $0.216 \times 1162 = 251$ B.t.u. Heating the air would take $0.40 \times 14 \times 0.24 \times (300-62) = 320$ B.t.u. The sum of these is 1152, which subtracted from 3168, the total heating value of tan with 60 per cent moisture, leaves a net value of 2016 instead of 1379, the figure given in the table. The efficiency would be $2016 \div 3168 = 63.6$ per cent, instead of 43.5 per cent, and the evaporation from and at 212 deg. $2016 \div 970 = 2.08$ lb. instead of 1.42 lb.

Prof. F. R. Hutton. In 1874, the late Robert H. Thurston presented a paper on The Efficiency of Furnaces Burning Wet Fuel,

before the American Society of Civil Engineers.[1] At that date few engineers were paying attention to fuel economy, and there was little widespread knowledge as to the details by which it would be obtained. There was of course no formulated code for boiler testing. This paper introduced the writer at that time to the problems of boiler testing, and recorded for the first time for him the formulæ for the barrel type of steam calorimeter.

2 The two furnaces examined were designed to meet the same requirements as are assumed in Mr. Myers' paper; but the press which may be expected to expel a proportion of the water absorbed from the leaching process was not in use, and no data were given as to the proportion between the dry bark ground at the mill and the weight of wet leached fuel delivered at the fire room. The Dutch oven type of furnace was in use, consisting of a fire-brick chamber covered with a reverberatory arched roof. The fuel was fed in at the top of the oven through two holes in the length of the grate. The grate was of fire brick moulded to obtain a semi-cylindrical surface to the upper and lower surface of each bar unit, the concave side being downward towards the ash pit. A large proportion of the finer tan lumps was expected to fall through the holes in the arched bars of the grate and complete their combustion there on the ash-pit floor.

3 But it is very plain from the results of the tests that the furnaces were on very much the same plane of efficiency as those reported by Mr. Myers, since the respective results of evaporation from and at 212 deg. per pound of combustible were for the Crockett furnace 4.41 lb., for the Thompson 5.68 lb. and for the Myers' furnace 5.43, 4.71 and 4.54 lb., if equal accuracy be assumed in the old test; as compared with the new. This is open to doubt, however, as certain figures were assumed or deducted from other experiments and were criticized in the discussion of the results.

4 The present paper is especially interesting to the writer, because it represents the work of a furnace designed by Mr. Myers which seems to incorporate some eminently sound principles. I think all will agree that the three cardinal principles for the complete and smokeless combustion of a reluctant fuel involve the following:

a Time enough for access of oxygen in the air to the carbon gas from the fuel.

b Temperature enough for the rapid and complete chemical union of this oxygen with carbon and hydrogen.

[1]Trans. Amer. Soc. C. E., No. 102, Vol. III, 1874, p 290.

c Room enough for each atom of fuel gas to meet the oxygen atoms with which it is to unite.

The practical attainment of these results is made more difficult when the fuel is wet and in small particles of light weight.

5 We have the conflicting conditions of a hot fire and a slow rate of combustion, to combine with an intensity of draft which shall not be high. Mr. Myers does this by using the step-grate idea, so as to admit the necessary air horizontally between the overlapping bars, whereby the dropping of fine fuel into the ash-pit is prevented: but in addition and as a special excellence of the design, the grate is made to consist of two sections facing each other with their planes parallel to the long axis of the Dutch oven and the shell of the boiler. They are, as it were, upon the inclines of a truncated capital letter V.

6 The bark is fed by a measuring stoker cylinder, which drops a determined volume upon the whole length of the upper bar at each partial revolution, and this fall of new material displaces downward some of what has been drying and growing ready to ignite from the previous charges. At the bottom of the truncated V is a dumping grate from which the residue of ash may be released at intervals.

7 The consequence of the inclination of the two grate sections, with a horizontal inflow of the air, seems to be the same as is produced in a successful form of burner for acetylene gas. The two currents of gas and draft appear to meet in the center or in the axis of the V and an intense combustion takes place there, the heat of which reverberates downward from the arch of the oven, raises the temperature of the upper layers of fuel, and stimulates the rate of the union of combustile with oxygen. Such a furnace of course is not subject to the alternations of the "famine and feast" conditions when excess of wet fuel deadens the fire and causes a smoky and slow combustion, alternating with high heat and good flame and followed in turn with burned-out spots in the fire until fresh charges came in through the holes.

8 The system is also most effective for "bagasse," the wet juicy fibre of the sugar cane after passing through the pressing rolls. This is more difficult to stoke mechanically than the comminuted bark, but the requirements for its successful combustion are very satisfactorily met. Sawdust and scrap from wood-working shops also are burned in furnaces of this design with less danger from sparks at the stack.

9 Referring to the summaries by the author, it should be plain that the geater surface must be reduced if the tan is press-treated

(Par. 56) to remove moisture. Bulk for bulk, there are more heat units per unit of volume or of weight after a volume or weight of water has been expelled than there were when the tan was saturated and not pressed. If the fire is hot enough to dissociate the oxygen and hydrogen which compose the water, the heat for such dissociation is drawn from somewhere: doubtless from the flaming gases, where the process takes place, and of course they are cooled, and perhaps killed.

9 If such oxygen and hydrogen recombine, nothing is lost, and perhaps a mechanic-thermal advantage is reaped because the hydrogen flame is longer than the carbon flame. If for any reason such dissociated hydrogen does not get a chance to recombine from lack of temperature or time or room, there is a loss. Mr. Myers' results should serve to check the claims still advanced at intervals, that the combustion of steam-gas is a source of any great possible economy.

The Author. Mr. Cary in his discussion has described the McMurray tan furnace—one of many different types and designs now in use. All the ordinary forms of tan furnaces feed the fuel through holes in the top or arch over the grate. The number and arrangement of these feed holes vary in the different designs, but they all form a bed of fuel composed of cones of tan. For this reason they are all subject to the objection made by Mr. Cary, i. e., that the fuel burns away most rapidly around the bottom of these cones where the depth of fuel is least. The central parts of the cones offer great resistance to the draft so that active combustion takes place on only a small percentage of the entire grate surface. This necessitates large grate surfaces and large furnaces with attendant radiation losses.

2 Another objection to the cone method of feeding the fuel, especially when only a single row of feed holes is employed, is that the fire is actually divided into a number of small fires around the bottom of the cones. This multiplicity of small fires separated by heaps of wet tan of low temperature results in lowering the furnace temperature and in retarding combustion.

3 In furnaces of this type with careless firing the writer has seen fully one-half of the grate surface doing no work at all in the way of any active combustion. These ill effects are best eliminated by very frequent feeding of the tan in small amounts, so that the percentage of wet tan in the furnace at any time is very small compared to the actively burning mass. High furnace temperature is thus maintained, more grate surface is active and the rate of combustion per square foot is greatly increased. The result is less grate surface required,

smaller radiation loss due to smaller furnaces and greater ease in handling and cleaning the fires.

4 In general the greater the number of feed holes the higher will be the rate of combustion and the smaller the furnace required. Rapid firing in small amounts to equal the rate of combustion in the furnace is productive of best efficiency with any of the usual types of tan furnaces.

5 Tan presses of different makes, but all of the same type described by Mr. Cary, have been experimented with by the writer. It was found that with careful adjustment and attendance the presses would equal the performance quoted by Mr. Cary but that under tannery conditions of indifferent attendance and unskilled labor the presses do not maintain their efficiency.

6 The interference of the steam gas evolved from the fuel with the union of the combustible gases with the oxygen must be overcome by providing large combustion space, preferably over the fuel bed, by special baffles or by special draft action as in the writer's design of automatic furnace shown in Figs. 2 and 3 and referred to by Professor Hutton.

7 The chemical composition of tan is assumed by Professor Kent to be practically the same as that given from an actual analysis in the author's paper. The heating value according to Dulong's formula is 7920 B.t.u. per lb., whereas the results of a large number of tests in a bomb calorimeter by Dr. Sherman, shows the heating value of a pound of dry hemlock tan to be close to an average of 9500 B.t.u.

8 I have carefully read the record of tests on tan burning furnaces made by Prof. R. H. Thurston, and presented in a paper before the Franklin Institute in 1874. Professor Kent states that some of the results there given are higher than those determined in recent practice by the writer. The two evaporative results by Thurston are given as 4.24 lb. equivalent evaporation from and at 212 deg., in the boiler per pound of combustible for the Thompson furnace, and 3.19 lb. for the Crockett furnace. The corresponding figure obtained by the writer in his automatic furnace was 5.55 lb.; that is, over 31 per cent better than Thurston's best result.

9 The writer finds that the evaporations of 5.68 and 4.41 for the Thompson and Crockett furnaces respectively were obtained by Thurston by *adding to the evaporation in the boiler the amount of moisture in the fuel evaporated from and at 212 deg.* A similar addition to the writer's evaporation in the boiler of 5.55 lb. would make an evaporation of 7.75 lb. including the moisture in the fuel. The latter

figure is therefore the one to be compared to Thurston's result of 5.68 lb. On the same basis of calculation the economic result of present best practice is over 36 per cent higher than the best result recorded by Thurston.

10 Moreover the highest result in the Thurston test was obtained by a rough volumetric approximation of the weight of the fuel used. It was not weighed to the fireman as in all the author's tests. Furthermore, both the weight and temperature of the feed water were merely approximated and assumed to be correct in the Thompson furnace test; whereas these values in the author's tests were all observed and recorded in a most accurate and systematic manner.

11 The accuracy and reliability of these old tests is very much to be doubted, as Professor Hutton suggests. But even if taken at their full values it is seen that the results of present practice have exceeded the old results by over 30 per cent.

12 Actually the present results are probably even higher than this, from a comparative standpoint, for the reason that in the old days of tanning, the moisture in the tan was less than in present practice. This consideration would have given the Thurston tests a decided advantage in the shape of a greater available heat value of the fuel. Thurston gives the moisture contents of the fuel as fired as 55 and 59 per cent, whereas the moisture in the writer's automatic furnace test was 65.3 per cent.

13 This increase in moisture is due to radical changes in the process of leaching the bark. Where formerly the bark was treated with cold, or nearly cold, water it now is leached at temperatures as near the boiling point as possible, and is subjected to the leaching process two or three times as long as in the former methods. This is on account of the high price of bark nowadays, which makes it pay to leach out as much of the tannin as is practically possible. Some tanneries to-day leach their bark so thoroughly that only $\frac{1}{2}$ per cent to 1 per cent of tannin remains in the spent tan.

14 The author desires to add that all results and data given in his paper are results of actual tests made under working conditions. No assumptions or theoretical calculations are involved in the conclusions. The feed water was in every case measured by means of two tanks or barrels set above a reservoir from which a separate feed pump supplied the boiler. Feed connections were so separated that it was physically impossible to pump the water elsewhere than in the boiler being tested. All connections involving a chance for leakage were blanked off. Valves were never assumed to be tight

but were proved so during the entire test by means of an open-tee arrangement which would show any leakage.

15 The temperature of water entering as well as leaving the measuring barrels was taken at frequent regular intervals. The barrels were calibrated by weighing when filled to their overflow pipes with water at the temperature which the feed water had averaged during the test.

16 The fuel was in every case weighed in equal amounts to the fireman. A sample corresponding to each 200 lb. was taken, kept in closed receptacles and at the end of the test was mixed, and quartered down to a quart or two quart sample which was sent in sealed jars to Dr. Sherman for determination of B. t. u. and moisture. All readings and observations were obtained with like regard for accuracy of results.

17 In Par. 3 Professor Hutton also compares the best results obtained by the writer with those of Professor Thurston; but as before pointed out, the results are on a very different basis and are not comparable, unless the moisture in the fuel is also added to the equivalent evaporation obtained in the boiler. If this is done the following table gives a correct comparison:

POUNDS EQUIVALENT EVAPORATION FROM AND AT 212 DEG.

Including Water In Fuel		Excluding Water in Fuel	
Thurston Tests	*Myers Test*	*Thurston Tests*	*Myers Tests*
5.68 for Thompson furnace	7.75 for Myers furnace	4.24 Thompson furnace	5.55 for Myers furnace
4.41 for Crockett furnace	6.63 for present ordinary furnace	3.19 for Crockett furnace	4.30 for present ordinary furnace

The table shows that when compared on the same basis of efficiency the art of tan burning has been greatly improved over the old methods, both with improved and ordinary furnaces.

18 Thermal efficiency is of course the safest and most accurate basis of comparing results of various boiler and furnace settings, and the highest result yet obtained in a reliable witnessed test in tan burning was 71.1 per cent. This is based on available heat in the fuel as fired after allowance is made for evaporating the moisture in the fuel. This test, which was made on the automatically stoked furnace before referred to, showed an efficiency of boiler and furnace of 54.4 per cent, based on the total heat of the fuel.

THE DESIGN OF CURVED MACHINE MEMBERS UNDER ECCENTRIC LOAD

By Prof. Walter Rautenstrauch, Published in The Journal for Mid-October

ABSTRACT OF PAPER

This paper is concerned with establishing a dependable method of procedure for the design of the principal sections of curved machine members, such as hooks, punch and shear frames, and the like. The basis for this method of design is the theory of the maximum straining action in hooks devised by E. S. Andrews and Prof. Karl Pearson of London University. Experimental results are submitted in support of the theory. Comparison is made of the maximum straining action predicted by the formula due to Unwin now in common use and by the analysis due to Mr. Andrews and Professor Pearson, with that found by experiments on hooks ranging from 2 tons to 30 tons rated capacity.

DISCUSSION

Prof. Gaetano Lanza. A careful perusal of the articles of Messrs. Pearson and Andrews in Drapers' Company Research Memoirs, containing the formulæ referred to by the author, reveals no flaw in the deduction of the formula for the greatest tensile stress at the section of greatest bending moment, provided it is regarded as a formula which gives the relation between the load on the hook and the tensile stress mentioned, and provided the section of greatest bending moment remains plane.

2 To determine in all cases, however, the relation between the load corresponding to a greatest stress at the above stated section, equal to the tensile elastic limit, and the elastic limit as determined by the methods of measurement employed, would, in my opinion, require a set of tests upon a series of hooks varying in their proportions to a much greater extent than those mentioned by Prof. Rautenstrauch, in which the formula of Prof. Pearson would make the two loads cited nearly equal. An example of such a case, in which this result does not hold, is a set of hooks tested under the direction of Prof. C. E.

Fuller, which were really open links of circular form, made by bending hot and annealing square bars, the side of the square being 0.75 in., where $\rho_0 = 3$ in., and where the load at the elastic limit, as determined in a similar manner to that pursued by Prof. Rautenstrauch, was 1100 lb.

3 For these hooks we should have $\gamma_1 = 1.0074$ and $\gamma_2 = 0.00658$.

4 The greatest tensile fibre stress at the section of greatest bending moment, if computed by the ordinary formula, would be 48,600 lb. per sq. in. and, if computed by the theory of Messrs. Andrews and Pearson, would be 59,300 lb. per sq. in., whereas the tensile elastic limit of the material was 30,000 lb. per sq. in.

5 In seeking an explanation of these apparently discordant facts the following observations should be kept in mind.

a In a straight beam we should naturally expect the elastic limit as determined by measuring deflections to be greater than that corresponding to a greatest fibre stress equal to the tensile elastic limit, the excess varying with the span.

b The methods used in all the experiments cited have been practically the measurement of deflections.

c The deflections, whether of beam or hook, cannot be determined by computation from the stresses at the section of greatest bending moment only, but depend also upon the stresses at the other sections.

d In the hooks tested by Prof. Rautenstrauch the section of the hook is a varying one in which the stresses at sections other than that of greatest bending moment have not been examined.

Hence it seems to me that before we can consider that a complete solution of this problem has been attained, we need

a A more extended series of tests which shall include a considerable number of hooks of each kind.

b An experimental determination, both for beams and hooks, of the relations between the elastic limit as determined by deflections and the load corresponding to greatest fibre stress equal to the tensile, or compressive elastic limits, in the case of varying spans and other proportions.

Chas. R. Gabriel. The results of tests of crane hooks and the figures obtained by the old and new formulæ, to which Professor Rautenstrauch calls attention, are very important as regards crane hooks and similar members of machines. If such members are not as strong as computed by the usual formula for combined bending and tension it is none too soon for engineers to be made acquainted with the fact. This is especially so because of the fact that metal beams of solid cross section, similar to the cross section of a crane hook when subjected to simple bending, show greater strength than that due to computation, at least when subjected to a breaking test. This excess of strength is so great, especially in beams of cast iron of certain cross sections, as to justify confidence in lesser dimensions for straight beam members than those that would be prescribed by calculation based on the tension and compression moments of beam sections. Similar excess over calculation, of ultimate breaking resistance by test, exists in shafts subjected to torsion.

2 One would naturally expect to find a similar excess of strength in crane hooks when put to test, but the results to which consideration is invited show quite the reverse to be the case, and are none the less valuable because disappointing.

3 As regards machine members, such as the overhung frames of presses, punching and shearing machines, etc., the large majority of such frames require to be rigid under their working loads, to an extent that renders them perfectly safe from failure by breaking. A great many points have to be considered with respect to dies being thrown out of line by the springing apart of the upper and lower arms of the frames. A small amount of such deflection would in some cases be sufficient to cause the shearing of expensive punches by the dies, rendering them unfit for the accurate work intended. In some few other cases, such as riveting, a comparatively large amount of deflection is permissible, and in some instances the proportions of a frame may be considered with respect to safety from rupture alone.

4 The cross sections of overhung frames must of necessity differ a great deal in different machines, also the relative amount of overhang or throat, depth of gap and general form of frame, whether curved similar to a crane hook, or extending straight up and down comparatively short or long distances. Various kinds of cross section such as solid rectangular, T, H, box or combination of box and rib, all have their appropriate uses. The successful designer has at times to depart considerably from formulæ that have been in use and must combine much practical judgment and observation in his work.

Factors of safety must vary from 3 to 50 or more, and stresses accordingly.

5 It is hardly to be expected that a formula for strength of crane hooks can be immediately applicable to all the various cases of overhung machine frames, but we judge it might be applicable to small frames of solid section and short overhang. Frames having a long overhang, such as represented by Fig. 1 in the paper, would in our opinion be a more trustworthy subject for the application of the useful bending formula than frames having relatively a much shorter overhang, such as indicated by the dimensions in Fig. 4. This is because the greater the overhang the more significant becomes the simple bending moment and the less significant the direct tension in the back of the frame.

6 Referring to Fig. 4, it is noticeable that the metal in the back of the frame is very thin. In frames where rigidity is the prime consideration, we believe it is a common error of designers when using cast iron to place too little material in the back. This no doubt arises from the known high compressive resistance of cast iron, without regard to its elasticity under compression; frames being designed accordingly, with regard to resistance to breaking rather than with regard to resistance to deflection. We have known of many cases where frames could be greatly stiffened by merely taking metal from the front web and putting it on the back web.

Prof. Wm. H. Burr. Professor Rautenstrauch has added a very interesting chapter to the lirerature of this subject, but there is perhaps a little more to the matter than has been indicated, and it bears a good deal upon what has been said by the last speaker. Doubtless the analysis based upon Professor Pearson's paper, as an analysis, is a decided improvement upon the Unwin formula, but again there comes in the same question raised in connection with reinforced-concrete beams. This analysis, whether by Professor Pearson or Professor Unwin, is based upon what is ordinarily known as the common theory of flexure, which belongs accurately only to straight beams of very small depth in comparison with the length.

2 Hooks and all such members as those shown by the author are exceedingly short as beams, and they are also curved. These conditions completely demoralize the analysis as based on the common theory of flexure, and it is not a matter of surprise that hooks should show so much greater carrying power than the computations would indicate. In fact, it is precisely in line with what we find in other short beams.

3 The pins at the panel points of pin-connected bridges are designed by the common theory of flexure. Yet if one should compute the extreme fibre stresses in those pins at some panel points as they have existed, they would be found to run up not only to 142,000 lb. per sq. in., but to 180,000 or 190,000 lb. in structural steel. A partial explanation lies in the fact that an analysis is used, which, strictly speaking, does not apply to these conditions. The hook and all such members, as well as bridge pins, are short, thick beams to which the usual theory of bending does not strictly apply.

4 Again, one will find that in bridge specifications, the regular working fibre stresses in pins are permitted to be at least 50 per cent greater than in the tension members of the truss; that is, one may have a working stress of perhaps 14,000 lb. in bars, and a fibre stress in tension of 18,000 or 20,000 lb., sometimes even 24,000 lb. in pins. This is due to a fact I have already mentioned, that as a matter of accurate analysis, the common theory of flexure should not be used in connection with such members; but there is nothing else to be done.

5 That again brings me back to the same point made in connection with concrete beams. The proper procedure is to settle upon some sensible working formula, just as we do in connection with the pins in bridges, make tests of such members, and deduce from these tests such empirical quantities as may be properly used in the formula, so as to make the results of the analysis in that way conform to safe and sensible practice.

George R. Henderson. That we get a rather greater strength than would be expected by the Unwin formula, especially in the case of hooks, agrees with my practical experience. A few years ago we purchased some 60-ton cranes, and when it came to the detail of the hook to lift the 60 tons, the design submitted by the manufacturers was for a hook smaller than we thought would be good practice to accept. We calculated to reduce the total strain due to the vertical stress and the bending moment to about 12,000 lb., which we considered would give a factor of safety of five with the material used. It was pointed out that the hook did not conform to the specifications, and that a larger hook was desired. These larger hooks were provided and they looked gigantic.

2 A little later the question came up again, when the manufacturers stood on their dignity and claimed that the hook was stronger than my calculations showed, and to confirm their case referred to

tests at the Watertown Arsenal, which we all consider pretty good authority. The hook tested was rated as a 20-ton hook, but it had been subjected to a weight of 162,000 lb., at which it merely bent but did not break.

3 These tests were to determine the ultimate strength, whereas the paper deals with the elastic limit; but practically, I think, the ultimate strength interests us as much as the elastic limit. By the regular Unwin formula, which has been somewhat condemned this evening, the stress per square inch in the hook, when weighted to 162,000 lb., at which it simply opened, would indicate 142,000 lb. per sq. in. fibre stress, which, of course, is absurd. So, from the actual tests, it is very evident that the hooks are considerably stronger than the Unwin formula could indicate. In discussing this matter with well known machinery builders, such as William Sellers & Company, we found that while the strain on the hooks might figure at 17,000 lb. per sq. in., from the formula, and show a factor of safety of only three, actually the factor of safety must have been five or six.

4 If possible, I would like to know how the author can reconcile these facts, compared with the practical ultimate strength tests, in connection with the elastic limit.

A. L. Campbell.[1] Table 2 of Professor Rautenstrauch's contribution shows an excellent agreement between actual test conditions and the results obtained by the formula which is the basis of his discussion.

2 A much simpler formula is used by the writer for similar computations. A crane hook or the frame for a punch is really a tension member with an exaggerated eccentric load. The maximum unit stress in such a tension member may be proved equal to

$$f_t = \frac{W}{A}\left(1 + \frac{l\,(1-e)}{R^2}\right)$$

using the author's notations. The radius of gyration, R, is equal to $\sqrt{\frac{I}{A}}$. Applying this formula to the frame shown in Fig. 4 gives $f_t = 7600$ lb. per sq. in. This stress is 90 per cent of that given by the more complex formula.

[1]The Solvay Process Co., Detroit, Mich.

FRANK I. ELLIS. While the paper, together with the article in the American Machinist to which it refers, covers very fully the design of hooks, giving results which agree remarkably with actual tests, its application to shear housings is not quite clear to us.

2 We note primarily, that in the derivation of his formula the writer has assumed the entire area to be in tension i. e., the neutral axis to lie entirely without the section. While this condition is almost universally correct in hooks, it will seldom be encountered in shear housings, but still it appears to have important bearing on the form of the equations.

FIG. 1 FRAME WITH INFINITE RADIUS

3 Another point which is not quite clear to us, but is a matter of great importance, is the assumption of the value of ρ, the radius of curvature of the gravity axis of the section. In the case of a hook, this of course is quite obvious, but in machine members, such as shear housings, this seems far from being the case. For instance, in a housing of the general form of sketch shown in Fig. 1 herewith, we would have an infinite value of ρ. This would reduce the formula to a case of simple tension, which is obviously incorrect, giving stresses that would be very much less than would be obtained by actual test. On the other hand, if we consider an extreme case as per Fig. 2, where the value of ρ is very small, the stress as calculated by the formula

would be very much in excess of what could possibly exist in the actual casting.

4 The example of a shear housing which Professor Rautenstrauch has chosen as an illustration appears to us to be at variance with our experience. The stress calculated by the new formula is almost three times that obtained by the usual methods of computation. In our experience, cast iron shear housings in which the calculated

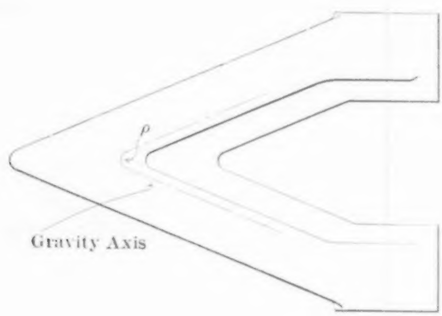


FIG. 2 FRAME WITH SMALL RADIUS

stress is 3,000 lb. per sq. in., never break except through defects in the casting, a condition which could hardly exist if the actual stress were in the vicinity of 9,000 lb.

5 In conclusion we may say, that aside from the seeming obscurity of the paper on the above points, we consider the formula to be of considerable value in the cases it is designed to cover. We regret that we have been unable to give it the time it deserves, and trust the above points will be made clear by the author.

E. J. LORING.[1] The results of the author show such striking discrepancies from the results by the usual methods of calculation that his analysis of the problem merits the most careful consideration. These results clearly show that the stresses and particularly the maximum stress in a curved piece under the combined direct and bending load to which such hooks and gap frames are subjected cannot properly be deduced from the simple combination of direct and bending stresses as determined by ordinary analysis from the stresses in a single plane, but may be influenced to a greater extent by con-

[1] Loring Speed Gauge Co., 76 Highland Ave., Somerville, Mass.

ditions outside of the section plane, such as the relations connecting that plane with those nearby on either side.

2 This difference between straight and curved members arises from a different distribution of stress due to the variation of length of fibres at different parts of the section as taken between similar adjacent sections.

3 The usual deduction for stress in straight members commonly applied to this problem assumes

a Planes remain planes after bending.
b Strain is proportional to the distance from the neutral axis.
c Stress is proportional to strain and therefore the stress is proportional to the distance from the neutral axis.

4 The assumption that stress is proportional to strain is true only as referring to unit strain, as long fibres will yield more under a given stress than shorter ones. In the case of straight members the adjacent minimum sections are parallel and the elementary fibres therefore all of equal length, and the assumption may be applied. In the case of a curved member, which I would define as one in which the locus of the centers of gravity of the minimum cross-sections is a curved line, these sections are not parallel, but radiate from a center of curvature so that the fibres are not of the same length throughout the section, and a correction must be made for the variation of the length of fibre before this assumption can be applied. This point has been generally overlooked or considered negligible, and in this point is to be found the explanation of the difference in results. I might add that this exemplifies the danger of applying a formula to conditions which it was not intended to represent.

5 I am not certain that I can agree with the author in the use of the theory of lateral contraction in the analysis. I cannot at this moment see why it is any more necessary in the case of the hooks tested than, for example, in the case of the test bars from which he deduced the fibre stresses. Taking only the common assumptions, with the correction for the length of fibre, as above noted, it is possible to obtain results in very close agreement with those given by the formula recommended by Professor Rautenstrauch. In place of the usual straight-line diagram of stress on the section these assumptions give the stress at any point as varying according to

$$\frac{y}{1+\frac{y}{\rho}} \quad \text{or} \quad \frac{y\rho}{\rho+y}$$

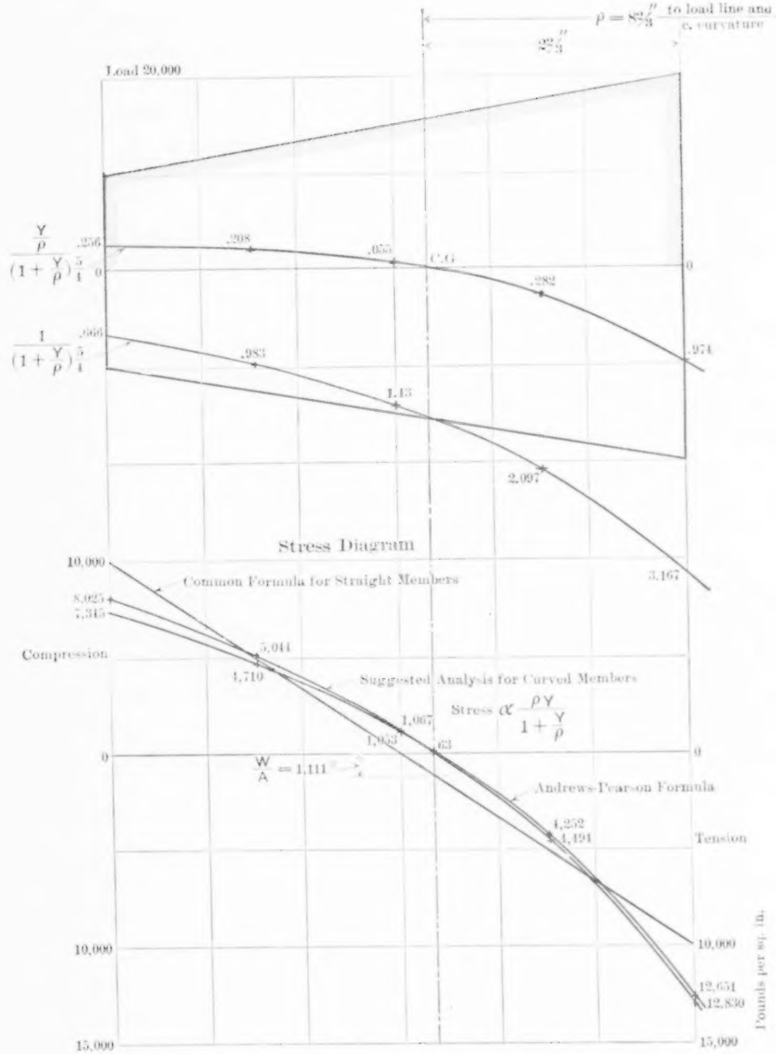


Fig. 1 Diagrams for Trapezoidal Sections of Straight and Curved Members for Equal Intensity of Stress

using the symbols of the paper, and from this may be determined this important fact; that for the case represented by the hooks, where the line of application of the load contains the center of curvature, the neutral axis contains the center of gravity. In other words, instead of the stress at the gravity axis being equal to the distributed stress as is true for straight members, the stress at this point in a member with this degree of curvature is zero, and this represents the manner in which the stress "piles up" toward the inner edge of a curved member. This condition of stress at the center of gravity would be represented in the analysis of the paper by the condition $\gamma_1 = 1 + \gamma_1$. The empirical formulæ recommended give $\gamma_1 = 1 + 1.1\ \gamma_2$ and the data on the hooks give a variation from $\gamma_1 = 1 + 1.17\ \gamma_2$ to $\gamma_1 = 1 + 0.88\ \gamma_2$ with an average of $\gamma_1 = 1 + 1.015\ \gamma_2$ so that it will be seen that this is approximately true by Professor Rautenstrauch's analysis; that the gravity axis is the neutral axis for this degree of curvature just as it is for transversely loaded beams.

6 I have applied the variation of stress

$$\frac{y}{1 + \frac{y}{\rho}}$$

given above to the solution of an assumed section and find that the stresses and their manner of variation are substantially identical, for this particular case at least, with those given by the author's method. I believe that an analysis can be made along this line that will give results very close to those shown and be more generally workable. The differential expressions for the net stress on the section and the moment of the stress are similar to those for a beam with the addition of a factor

$$\frac{1}{1 + \frac{y}{\rho}}$$

$$W \text{ varies as } \Sigma \frac{y\,\mathrm{d}A}{1 + \frac{y}{\rho}} \qquad Wl \text{ varies as } \Sigma \frac{y^2\,\mathrm{d}A}{1 + \frac{y}{\rho}}$$

It may perhaps be possible to deduce some general expression to be used as a factor of correction for curvature to be used with the usual methods.

7 The effect of the curvature is less, the greater the ratio of radius of curvature to the depth of section. In the case of hooks this means greater strength where the contour of the inner edge is elliptical instead of circular, so that the curvature at the most strained section is less. As the curvature tends to "pile up" the stress toward the inner edge, greater strength may be had by giving the hook a closer approximation to a Tee section, by which means the metal is massed better where the stresses are abnormally high. It would also appear that a high gap is stronger than a low one for the same depth, since a lesser degree of curvature is possible.

8 I must disagree with the statements in Par. 2 except as limited to curved members; also with the statement in Par. 8 that γ_1 and γ_2 are constants for all sections of similar form, except it be modified to say "of similar form, curvature and load distance." I question the significance of the quantity k in Par. 11, where it is stated to be the radius of gyration, as the value of k given for the punch frame following is the radius of gyration squared, or $\frac{I}{A}$.

9 In determining the maximum stress by the method which the author has proposed, the function γ_2 is the most important factor, and this function is obtained from the difference in area of two derived curves; the difference is small and the less the difference the greater the maximum stress. It would seem that there is great opportunity for inaccuracy in determining this factor. It also appears to me to be simpler to take

$$\gamma_2 = \frac{-\Sigma\left(\dfrac{1}{1+\dfrac{y}{\rho}}\right)^{\frac{3}{4}}\dfrac{y}{\rho}\,\mathrm{d}A}{A}$$

as originally stated for the purpose of the computation, rather than to use the value derived from it, of

$$\gamma_2 = \gamma_1 - \frac{\Sigma\left(1+\dfrac{y}{\rho}\right)^{\frac{1}{4}}\mathrm{d}A}{A}$$

for the reason that having the quantities for the determination of γ_1 for various points of the section, that is, the values of $\left(1+\dfrac{y}{\rho}\right)^{\frac{3}{4}}$

it will be simpler merely to multiply these by the respective distances from the gravity axis, plot the curve and integrate for the net area, rather than to proceed by raising the denominator to a new power and passing through all the processes anew.

10 It is stated that the standard section selected for the computation of constants for the empirical formula is not the most economic from the standpoint of equal tension and compression stresses. This is true even if the member is straight, in which case, considering the trapezoid only and omitting the curved ends, the maximum stress in compression is 85 per cent of the maximum stress in tension. All other parts remaining the same, for equal intensities of stress in the edges for a straight member, the half width of the narrow edge should be 0.095 r, as may be very readily demonstrated. The geometrical relations for the correct proportions of a trapezoidal section for equal intensity of stress in a straight member are so exceedingly simple that I want to give them here, particularly since so far as I know they have never been published. This relation is that the sides extended intersect at a distance from the far or narrow edge equal to the distance of the load line from the near or wide edge, and for the solution of this case we have

$$d = \sqrt[3]{6\,ky\frac{F}{f}}$$

where d = depth of section
y = distance of load line from the near edge
F = load
f = maximum stress at the near edge or far edge (equal)
and k is a design constant = ratio of depth of section to width of far edge.

11 For the case of equal stresses in a curved member of trapezoidal section with center of curvature on the load line, a similar relation may be deduced from the analysis that I have here suggested, but is not quite so simple: the point of intersection of the sides is given by the following construction. Lay off on the axis of symmetry and toward the far edge a distance from the *near* edge equal to the distance from the near edge to the center of curvature and load line. If this distance is greater than the depth of section, equal stresses may be had. If this distance is equal to the depth of section; i. e., if the point thus laid off is on the far edge, equal stresses require a triangle with this point as the apex. If the point is beyond the far edge,

divide the distance to that edge in thirds; then the stresses are equal when the sides extended intersect at the nearer point of division, one third of this distance from the far edge.

12 It will be noticed that this construction gives the radius of curvature for this limiting case[1] equal to 1.33 times the depth of section[1] instead of 1.75 as given by Professor Pearson. I have investigated this case for both degrees of curvature by the method involving the lateral contraction and find that using the formulæ given by Professor Rautenstrauch the curvature of 1.33 times the depth, measured to the gravity axis, gives a stress on the inner edge of 1.091 times that on the far edge. A similar operation for curvature of 1.75 as recommended by Professor Pearson, by his own method gives by my computations a ratio of stress of 0.912. A sharp triangular section such as this is however of little or no importance in actual construction, and the method of determining the proportions which I have given will, I think, be found to be of much more general application. I am unable to state at the present time whether a section having equal intensity of stress on the two edges is or is not the most economical of material; but presumably it is.[2]

Prof. C. E. Houghton. The agreement between the elastic limit as calculated by the proposed formula and that as derived from the tests, is to say the least, wonderfully close, and the wide variation between the experimental values and those calculated by the use of a theory that has been in common use for many years leads one to ask "Why are there not more failures in crane hooks?"

2 Objection has been made to the tests because the hooks were not loaded beyond the elastic limit. This seems to the writer to be a mistake. What the engineer is mostly interested in is the effect of loads that produce stresses within the elastic limit, since the great majority of the formulæ used for the calculation of stresses are based on theory that no longer holds true after the elastic limit has been exceeded.

3 Professor Burr has pointed out that the simple theory of flexure does not apply to curved members and Mr. Gabriel notes that stiffness and not strength is the controlling factor in many of the open-side machine frames. May not the fact that cast iron is used in the majority of such frames be another reason why the flexure formulæ cannot be expected to give correct results? The well-known fact

[1]Or depth of section equal to gap depth.

[2]Since writing the foregoing, Mr. Loring has found that the method suggested by him for the determination of the stresses—or a very similar one—is given in some detail in Hütte, from some German source dating 1902.

that the physical properties of any cast iron vary with the rate of cooling, and that the tensile strength and modulus of elasticity are not constant at all depths from the surface of any cast iron member, but vary throughout any given section, leads one to ask "Is it not more reasonable to use the simpler formulæ in the calculations for strength and to provide against possible errors by that useful and elastic term—the factor of safety?"

H. Gansslen.[1] The author's tests prove the correctness of Andrews and Pearson's new formula for figuring crane and coupling hooks. All the experimenters, however, seem to have limited themselves to these hooks, for which the formula appears to have been gotten out. Hook's law of the direct proportionality between stresses and strains also underlies the new formula and the fact that this law holds practically good on wrought iron, steel and similar materials would to some extent explain the good agreement of the results of tests and calculations by means of the new formula.

2 The author points out that the formula is applicable to punch and riveter frames. To generalize thus I believe is hardly wise at present, as all the various formulæ for figuring curved beams are more or less empirical and each of them is naturally proved to be true for a certain limited field of calculations only. Hook's law does not hold true for copper, cast iron, bronze, stones artificial and natural, etc., and this law giving the modulus of elasticity as constant is the basis of the formula.

3 Engineers know that the old formula for figuring a curved member in the same way as a straight beam gives too small factors of safety, but that we are now under-estimating the stresses in the throat of punch press frames $8500 \div 2450 = 3\frac{1}{2}$ times is surely saying much.

4 However, there is no use disputing the new formula in so far as tests have verified it and it is to be hoped that the author will have the opportunity of entering other fields of research besides that of crane hooks, and that of press frames would be a desirable one.

5 I have not come across a case where a punch press frame figured in the usual, but wrong way could have been $3\frac{1}{2}$ times under-estimated, roughly considered, by comparing the pressure exerted with the general behavior of the frame.

6 The old theory of flexure as applied to and compared with tests of cast iron has shown its inapplicability and this should make us all

[1] Mechanical Engineer, 404 Fisher Building, Chicago.

the more cautious in adopting the new formula for cast iron press frames before having the results of tests on hand that would justify us in so doing.

John S. Myers.[1] The author's presentation on the design of curved machine members and his article in the American Machinist of October 7, 1909 dealing exclusively with crane hooks, seem to indicate that the new theory is applicable to punch and riveter frames of the type shown in Fig. 1, where the throat is semi-circular, being struck with a radius having its center at O. In order, however, to find the radius of curvature of the gravity axis of the principal section it would seem necessary to plot points such as A, B, C, D, E, draw a curve through them, then, by trial, find the center O' of a circular arc which will pass through C and most nearly fit the curve for points intermediate between B and D.

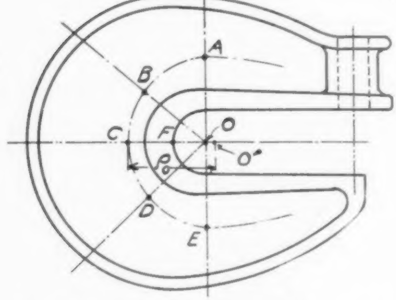


Fig. 1 Frame with Semi-Circular Throat

Curve $ABCD$ represents the gravity axis of the section. Point O is the center of the throat radius. Point O' is the center of a circular arc which approximately coincides with the gravity-axis curve for points between B and D.

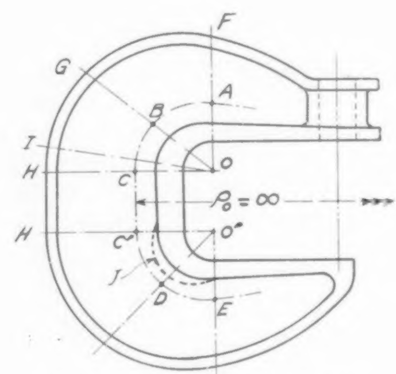


Fig. 2 Frame with Wider Gap than Fig 1

Curve $ABCC'DE$ represents the gravity axis. Between points C and C' this curve becomes a straight line; hence $\rho_0 = \infty$

2 If the above is consistent with the assumptions upon which the theory is based, it will be seen that the point O' is not necessarily coincident with O, and that to find the value of ρ_0 a layout must be made and the gravity axis of several sections determined. It

[1] John S. Myers, 2456 Almond St., Philadelphia.

is also seen that ρ_0 is not strictly a function of the throat radius nor is it equal to $OF + CF$ as one would at first suppose. This adds more complication to the problem, which is already vexatious.

3 Again, such frames are not always made with the throat struck with a single radius; in fact, this is the exception rather than the rule for quite a large class of machines, which have a wider "gap" to accommodate the work and are more like that shown in Fig. 2. Here the curve representing the gravity axis is a straight line between points C and C', in consequence of which $\rho_0 = O, O$ and it would therefore seem that the new theory did not apply to this portion of the frame. Now, if this be the case, and we design that portion of the

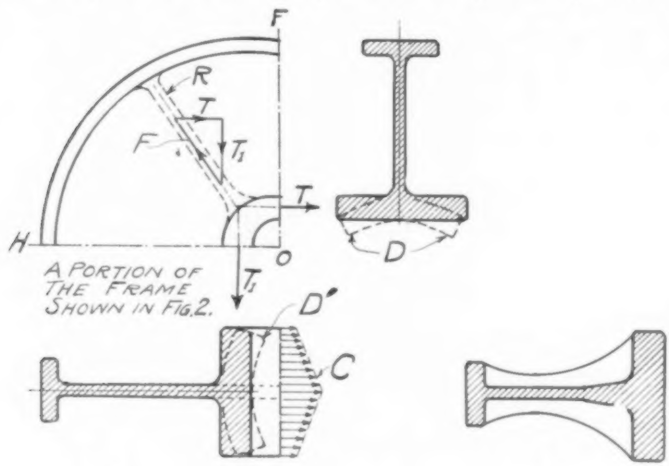


FIG. 3 FIG. 3a

FIG. 3 SHOWING HOW THE RAPID TRANSITION OF STRESSES INDUCES LOCAL STRESSES. FIG 3a. PROPOSED SECTION

frame between OH and $O'H'$ according to the old theory of straight beams, but design section OI according to the theory of curved beams under discussion, it would appear from an inspection of the results given by Professor Rautenstrauch that section OI should have about three or four times the flange area of section OH. Of course the metal at the corners could be thickened, as indicated by the dotted line at J, but it would be out of the question to double or treble the usual flange thickness, which is what the new theory seems to indicate as necessary.

4 It would be very interesting to know how the new theory could be properly applied in such a case; whether, for instance, it is entirely applicable at the section OG but gradually merges into the old theory

at sections OF and OH; or whether it has not, as yet, been sufficiently developed to be generally applicable to sections other than those at right angles to the line of action of the force.

5 Generally speaking, a structural engineer never puts in curved tension or compression members because he knows that force either travels in straight lines or else produces bending strains; but the average designer of machinery seems to delight in curved ribs, bent levers, and the like. The average mechanical draftsman makes layouts as if he held the opinion that force travels along a curved rib in a manner somewhat similar to water flowing in a pipe and that it

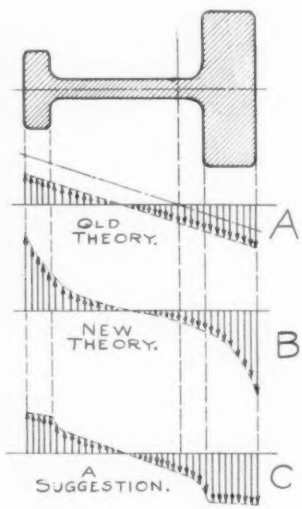


Fig. 4 Distribution of Stresses Under Different Theories

will, therefore, follow any devious or sinuous course in which he may choose to distribute the metals. Most C-frames seem to be designed on the foregoing assumption and, while it is an exceedingly difficult piece of mental gymnastics to follow the mathematics of the new theory, it is, however, quite easy to see that there are stresses induced in curved ribs which are usually ignored.

6 To illustrate the foregoing, Fig. 3 shows that portion of the frame of Fig. 2 which lies between lines OF and OH. Now, let T and T_1 represent the total tensions in the flanges on sections OF and OH respectively. By combining T and T_1 graphically it is seen that a resultant force F must, in some manner, be supplied to establish equilibrium. The most direct way of supplying such a force is by

the addition of a rib as indicated by the dotted lines at R which will distribute part of F into the web and deliver part of the force at the compression flange where there is a smaller, opposing resultant force. In the absence of any such rib the necessary force must be supplied by the web, partly through a local bending and distortion of the flanges as indicated by the dotted lines at D and D' and partly by a concentration of stresses towards the central portion of the flanges as indicated at C, this concentration being a direct result of the deformation at D'.

7 In supplying a rib R, if it was intended to carry the entire force F it would be necessary to make it about $1\frac{3}{8}$ times the average thickness of the flanges, but since the web can readily take half, or more than half, of the load it would seem that a rib of $\frac{5}{8}$ or $\frac{1}{2}$ of the flange thickness, narrowed down at the center as shown in Fig. 3a would be entirely sufficient, especially if the web be judiciously thickened and liberal fillets used.

8 As I understand the new theory it does not recognize any such concentration of stresses as indicated at C in Fig. 3 but, on the contrary, assumes a more rapid concentration towards the extreme fibres in a manner somewhat similar to that shown at B in Fig. 4. Now in view of the close accord between the new theory and the results of Professor Rautenstrauch's experiments, I am quite ready to believe that diagram A represents quite closely the actual conditions for straight beams of solid section, and that diagram B represents the most plausible theory for curved beams of solid section; but that for beams composed of heavy flanges and a light web the probable distribution of stresses is more nearly like that suggested by diagram C, and that so far as the curved form of the beam is concerned, it is not the curve of the neutral axis we are interested in but the curve of the flanges, and that this results in local bending and concentration of the stresses as already pointed out.

9 I have no well formulated theory to advance in explanation of my belief in a distribution of stresses like that indicated by diagram C but have sufficient faith in it to calculate sections of this nature by the very simple process of considering the stress to be uniformly distributed over the flange area and entirely neglecting the web; then at points where there is rapid transition of stresses, supplying ribs, thickening up the web and allowing a lower flange stress and liberal fillets. This procedure may sound crude to a scientific man, but it has, at least, ease of application in its favor and may yet be shown to be actually more scientific than the more laborious methods

usually pursued. As yet, I have not had the temerity to apply this method to large work but would like to have the opinion of those who have had experience along these lines.

The discussion concluded with an interesting talk by Carl G. Barth illustrated by a blueprint and blackboard sketches. Mr. Barth has not been able to prepare this for publication. Editor.

THE AUTHOR. The test reported by Professor Lanza is interesting, but I do not feel justified in replying without a review of the entire data on the experiment. The point made by him in Par. 5 in regard to deflections, is somewhat misleading. I did not propose in my experiments to determine the relation of total deflections to the maximum stress in the hook, but rather to find the load at which the total deflection ceased to follow the straight-line law. Since the total deflection is dependent on the deflection of all the sections, it is rational to suppose that when any variations occur they are due to the fact that the "fibres" in the most strained section have been stressed beyond the elastic limit. This is all we wish to know. The most strained section is without doubt the main horizontal section. The examination of the bending moments in other sections is of no value in these determinations.

2 Referring to Mr. Gabriel's remarks: I regret that so many designers persist in applying the formulae for determining maximum intensity of stress beyond their limits of application. No computations can be made to determine *ultimate breaking strength* and I see no reason why anyone should be surprised that there is a disagreement between the "results of computations" and the results of test. I did not choose to consider the matter of rigidity, which the title of the paper would lead one to believe should be included. Rigidity is, of course, a controlling factor in die work. The dimension of the metal in the back of the frame shown in Fig. 4 of the paper should be $1\frac{1}{4}$ in.

3 Mr. Henderson's remarks that his practical experience with hooks leads him to believe that a rather greater strength exists than can be expected from the Unwin formula, qualified by his report of certain tests, would lead one to believe that he has made use of Unwin's formula outside of its field of application. Unwin's formula indicates nothing beyond the elastic limit. There exists no method of analysis which enables us to determine the relation between the load on the hook and the resulting maximum intensity of stress

when that stress is beyond the elastic limit of the material. In reply to the statement that "the ultimate strength interests us just as much as the elastic limit," I would say that I believe designers will be treading on much safer ground when they confine themselves to proportioning parts with a factor of safety based on the elastic limit rather than the ultimate strength.

4 Mr. Ellis says in the second paragraph "We note primarily that in the derivation of his formula the writer has assumed the entire area to be in tension. i. e. the neutral axis to lie entirely without the section. While this condition is almost universally correct in hooks it will seldom be encountered in shear housings." No such assumption is made, nor is it universally correct in hooks. I believe that Mr. Ellis is also mistaken in his remarks on the particular form of the equation when ρ is infinite. When ρ is infinite the case is not that of simple tension but rather as expressed by Unwin's formula.

5 Mr. Loring's explanation of the two analyses, I regret to say is incorrect. Both analyses are founded on a determination of the relation between *unit* stretch and intensity of stress, but the real difference is found in the methods of evaluating the unit stretch. The older formula gives the *unit* stretch as

$$\lambda_y = \lambda_\beta + \frac{y'}{\rho'}$$

while the newer analysis gives

$$\lambda_y = \lambda_\beta + \frac{\dfrac{y'}{\rho'} - \dfrac{y_0}{\rho_0}}{1 + \dfrac{y_0}{\rho_0}}$$

where

λ_y = unit stretch of any fiber a distance y^1 from the gravity axis.
λ_β = unit stretch at gravity axis.
ρ' = radius of curvature at gravity axis after stretching.
ρ_0 = same before stretching.
y_0 = modified y' after stretching.

The newer analysis retains terms of the same order of magnitude as λ_y and therein lies the difference. The theory of lateral contraction is rationally applied in this analysis, its application being unnecessary to the test piece, since direct measurement of stress is made.

6 Par. 2 in the paper is obviously limited to curved members. The similar form referred to in Par. 8, includes the radius of curvature.

7 In Fig. 4, $k^2 = 68.56$. In Par. 13 the equation should be $\gamma_2 = \frac{k\,e}{0.7\rho^2}$. The method uesd for determining γ_1 and γ_2, I believe will be found more convenient than those proposed by Mr. Loring.

8 Professor Houghton will agree with me that a more correct analysis for straining action will permit a more intelligent use of the factor of safety.

9 Mr. Myers is quite correct in his remarks on the value of ρ_0. The analysis, however, does apply to the case of straight beams where $\rho_0 = \infty$, for which case it reduces to the form of the Unwin formula. The formula has not as yet been sufficiently developed to determine its usefulness in establishing proportions for other than those sections at right angles to the load. The difficulty of determining the stretch on sections at an angle to the load will leave this problem unsolved for some time. It is, however, rational to suppose that the flange on oblique sections should be thickened, but to what extent has not yet been determined. In regard to the behavior of a T-section, I would state that Professor Pearson has found experimentally that it is subjected to the same laws as a solid section. This indicates that the suggestion of Mr. Myers in Fig. 4 can hardly be accepted.

10 I judge from Professor Burr's remarks that he discredits the analysis by Professor Pearson on the basis that it is founded on the common theory of flexure, that is, it is not applicable to beams of very great depth compared with the length. I believe that if Professor Burr had given more thought to the matter he would not have made this statement. In view of the experimental results obtained by myself and others in verification of the theory and the lack of any data in verification of Professor Burr's statement, I am still inclined to believe that Professor Pearson's analysis is correct.

VENTURI TESTS FOR BOILER FEED

By C. M. Allen, Published in the Journal for Mid-October

ABSTRACT OF PAPER

The object of these tests with the venturi meter was to determine how well adapted it would be for use in measuring the feed to a boiler, in view of the variety of conditions under which it might have to operate. The methods of pumping the water through the meter, the different temperatures of the water pumped, various and fluctuating pressures and velocities of flow,—any one or several of these conditions might be met with in actual service, and the results obtained indicate that such occurrence would have practically no effect on the satisfactory performance of the work of the meter.

However, there are limits to this satisfactory operation of any one meter, and the lower limit for this size seems to be reached when the velocity of the water through its throat becomes lower than about 10 ft. per sec. In case the desired amount of water is smaller than the quantity which would produce this velocity in the meter, a smaller meter would be installed. It is evident from these tests that the venturi meter is sufficiently accurate for the majority of commercial or engineering requirements.

DISCUSSION

F. N. Connet. I think that the coefficients shown by Fig. 2 are slightly less than they ought to be, because I suspect that the "venturi head" was considered equal to 13.6 times the difference of the mercury levels in the manometer, whereas this ratio is actually 12.6. Correcting for this difference would slightly increase the coefficient and would make it correspond more closely to those obtained in our own experiments.

2 The correction necessary for difference in temperatures is not as great as with mechanical meters, for the reason that the venturi meter itself automatically compensates for one-half of the difference in specific gravity. In other words, if the water be hot and the specific gravity 2 per cent less than that for which the meter was calibrated, a correction of 1 per cent is automatically made by the meter and therefore a further correction of only 1 per cent is necessary, whereas, with a mechanical meter depending upon volumes, a correction of 2 per cent would have to be made if the readings were desired in pounds. The

reason for this difference between the two types of meters is that the flow through the venturi meter is proportional to the square root of the venturi head and is not directly proportional to it.

3 The only reason for not always obtaining accurate results with the venturi meter for boiler feed is the presence of severe pulsations in velocity due to the action of the feed pump. The most accurate results can be obtained when the feed pumps are of the centrifugal type and many such pumps of the two-stage or three-stage turbine variety are now in successful use. The pulsations which are due to the action of the water plungers or to defective valve action in a reciprocating pump, make it necessary to place a rather large air chamber directly on the pump, or on the feed line as close as possible to the pump. If placed on the feed line, it should not be connected on a tee set in the line but it should be so arranged that all of the water will pass through it. Furthermore, the cross section of such an air chamber should be large and the arrangement should be such that the surface of the water will rise and fall with each stroke of the pump, There should be a gage glass on the side of the air chamber so as to insure the presence of a sufficient vacuum of air. These precautions will render the velocity of the water sufficiently uniform to obtain accurate results with the venturi meter.

4 I notice also that the results obtained were not very satisfactory when pulsations were present and when the throat velocity was less than 10 ft. per sec. There were three reasons for this which seldom if ever exist in actual venturi meter installations:

a The instrument used in the test was a mercury U-tube or manometer, containing but little more than a pound of mercury. The inertia of the mercury was therefore small and the mercury levels were unsteady. In an actual installation a registering instrument is generally used which contains almost 100 lb. of mercury, the mere inertia of which has a decided "damping" effect.

b The graduations on a manometer scale are quite close together at low throat velocities. At 10 ft. per sec. throat velocity, the difference of mercury levels is only $1\frac{1}{2}$ in. In the registering instruments the movements are increased by a lever so that accurate readings are facilitated.

c During the tests described in the paper, the globe valves in the two pressure pipes were partially closed to minimize the mercury level fluctuations, and in all probability the valve discs were slightly loose on the valve stems. This therefore allowed the discs to behave like check valves

and permitted a freer flow in one direction than in the other; consequently incorrect mercury levels would result.

5 There are at least three better ways to throttle one or both of the pressure pipes than by using globe valves. The first and perhaps the best way is to use a capillary tube, say ¼ in. inside diameter by two or three feet long. The second way is to use a needle valve which is similar to a globe valve, but without a loose valve disc and with a long tapered point directly on the valve stem. The third way is to use a cock instead of a valve. Any of these methods of throttling combined with an ample air chamber permits accurate venturi meter readings at throat velocities as low as 2.8 ft. per sec. This extends the range of the meter from its maximum capacity down to one-thirteenth of the maximum.

6 Although a manometer, because of its portability and simplicity, is particularly well adapted to the making of short boiler tests, it nevertheless is not automatic and it shows the rate of flow only at the moment of observation, and if this rate fluctuates considerably from minute to minute, it becomes necessary to take very frequent readings. For this reason an instrument has been perfected which has two dials, one for indicating the rate of flow and the other for continuously recording this fate upon a circular chart paper. A special planimeter enables the charts to be measured so as to obtain the total quantity of water. This planimeter multiplies the factor of velocity by the factor of time and the product, of course, represents quantity. This type of recording instrument is largely used for meters 4 in. and smaller in diameter but for larger size meters the users generally prefer a three-dial instrument of the integrating type in order that the total quantity of water may be read directly upon a revolution counter without the aid of the planimeter.

Clemens Herschel. Professor Allen's paper shows, by tests properly and skillfully made, that the meter is reliable for hot water and boiler-feed service, and is new and unique as reproducing in tests of the meter the curious conditions to which a boiler-feed water meter is subjected. But for this feature the tests would have been only a repetition of other tests already made. Not that such repetitions are not desirable, especially when made as accurately and with the scope and purpose of those given in Professor Allen's paper. Further series of rests on venturi meters of all sizes, are in fact still called for in the interests of exactitude. But they can only in a general way confirm, not discover.

2 The point to be considered is, that several thousand venturi water meters are now in use, the world over. They are the embodiment of the action of one of the laws of nature, and are but little dependent on a correction by coefficients. They have been tested in various sizes, from ¼-in. to 10-ft. main pipe diameter, and operate exactly alike in all these sizes. They are also used to meter gases, brine and chemicals, and, as we see from the paper, to meter hot water. It is indeed a curious circumstance, that while the inventor and the manufacturers of the venturi water meter never expected to see many of these meters of less than 12 in. diameter used in practice yet the demand for hot-water boiler-feed meters has exceeded in value that of all the other sizes, for certain periods.

Sanford A. Moss. I understand from Par. 7 that the discharge of the venturi meter was figured on the basis of cold water with standard density in all cases, and that the theoretical effect of change of density was not taken into account in the formula. This would mean that Professor Allen's curve takes account of the effect of density changes, as well as all other changes. The actual formula used, and a sample of the calculations, might be a desirable addition to the paper.

2 Assuming that the above interpretation is correct, Professor Allen's curve shows that the actual flow in pounds per hour, with a given pressure, increases as the density decreases, due to rise of temperature. Is this not surprising? Theoretically, flow should decrease with the square root of the density. Of course change in the orifice friction coefficient, due to change of density, temperature, etc., might occur to such a great extent as to overbalance effect of density change. The actual orifice friction coefficient would then have a greater upward slope than in the chart so as to be over 98 per cent at 200 deg. Orifice friction coefficients for all density conditions and all fluids are usually the same for velocities occurring in practice, which are always above the "critical velocity" where fluid adjacent to a wall is stationary and where viscosity is a factor. Thus the orifice coefficient for air is the same as for water, even though the density is decreased about 800 times.

F. N. Connet. If I understand Dr. Moss correctly, he states that the quantity decreases as the density increases. With the venturi meter this depends upon the character of the graduations. If the units are cubic feet the readings *decrease* in proportion to the square

root of the increase of density, but if the units are pounds the readings *increase* in proportion to the square root of the increase of density. One is exactly the reverse of the other.

Geo. A. Orrok. I note that Professor Allen has obtained results for the coefficient of the venturi meter similar to those given by Clemens Herschel in his paper presented before the American Society of Civil Engineers, December 21, 1887, the lower values of the coefficient appearing at a velocity of about ten feet per second.

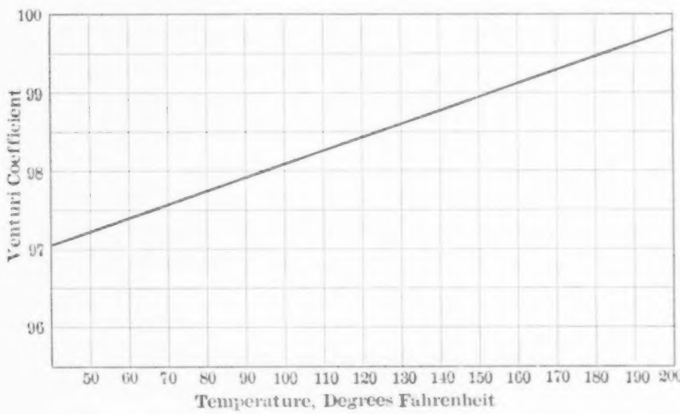
2 The New York Edison Company for some years has been using venturi meters for the measurement of water. We find them accurate and very convenient. For the last three years we have been using them in the testing of our boilers, having conducted a series of check experiments to determine the variations with temperature. Our condition is considerably better than Professor Allen's, since we use centrifugal feed pumps and consequently have a steady reading on the manometer.

3 In cases where we have both weighed and measured the feed water our results were remarkably close. On a 7-hr. test, where about 170,000 lb. of water was fed to the boiler, the meter exceeded the weighing by 631 lb., or approximately 0.37 of one per cent. In another test, in which nearly 200,000 lb. was fed, the difference was about 0.47 of one per cent. I believe the meter readings are more nearly correct than the weighing, as there was considerable opportunity for evaporation from the tanks in which the weighing was done.

The Author. Mr. Connet is correct in his statement concerning the coefficients of the venturi meter, relative to temperatures as shown in Fig. 2. There should be a correction. Each coefficient should be multiplied by the factor 103.8. This raises the coefficient to much nearer unity. The curve herewith shows the relation between the values of the coefficient for the varying temperatures.

2 I agree with Mr. Connet in regard to the throttling of the water in the pipes leading to the manometer. I believe the needle valve, or a fairly long pipe of small diameter, would be a decided improvement over the globe valves which were used in these experiments. We had not discovered that the movement of the end of the globe valves affected the reading, but Mr. Connet has had a good deal more experience along these particular lines, and I am perfectly willing to believe that this is true and that these fluctuations could be materially cut down and yet give the true mechanical average. This is

what we are looking for, and it is a good deal better than using maximum and minimum readings and then obtaining the arithmetical average. The mechanical average obtained by means of throttling is certainly more accurate because we do not know how long the maximum deflection continues, relative to the minimum.



Curve Showing Variation of Venturi Coefficient with Rise in Temperature

3 For the benefit of Mr. Moss, I would state that the density at different temperatures was considered. The following is a sample test giving an idea as to how computations were made:

If W = actual weight of water from weighing tank, then

$$W = 60\, w\, a\, C\, t \sqrt{2gh}$$

w = weight per cu. ft. at the temperature

a = area venturi throat

C = venturi coefficient

t = time in minutes

h = venturi head

$$C = \frac{W}{60\ w\ a\ t \sqrt{2\,gh}}$$

$$C = \frac{W}{1.48\ w\ t \sqrt{h}}$$

DATA OF TEST

Time 3:40 – 3:51; duration 11 minutes.	lbs.
Weight of tanks at beginning	1158
Weight of tanks at end	5369
	4211
Deduct for tank calibration.	20
	4191
Add for evaporation	2
Total water	4193
Mean mercury deflection	17.24 in.
$h = 1.05 \times 17.24$	$=18.1$ ft.
$\sqrt{h}$	$=4.25$

w for temperature of 137 = 61.43

Weight = $1.48 \times 11 \times 61.43 \times 4.25 = 4250$

$C = \dfrac{4193}{4250} = 0.986$ coefficient of venturi meter.

COOLING TOWERS

By J. R. Bibbins, Published in the Journal for Mid-November

ABSTRACT OF PAPER

Developments of recent years of both steam and power plants have demonstrated the usefulness of the cooling tower. This piece of auxiliary apparatus has always been more or less neglected.

It is true that in some plants the maximum effectiveness of the cooling tower and that of the condensing plant are in a sense diametrically opposed—one profits by the shortcomings of the other. The tower works best when the vacuum is lowest. On the other hand this tends to a general operative equilibrium and often saves the day when two interdependent types of equipment would succumb. Fortunately, improvements in condensers is being actively pushed, the trend being to secure higher hot-well temperatures. This immediately enhances the effectiveness of the cooling tower. Similarly, in gas-power plants, the possibility of cooling jacket water by means of this apparatus is favored by the high temperatures of discharge which prevail in engines of good construction. It is not an impossible state of affairs for the jacket water in a gas-power plant to cost more than the fuel, if not cooled and used over again, so that from all standpoints the cooling tower is worthy of careful study.

It is the object of this paper to bring into concrete form for discussion the most prevalent ideas in cooling tower construction, and a simple, inexpensive type employing lath mats is suggested together with suggestions for a combination of natural draft and forced draft types.

The performance data included in the paper are merely to give some idea of the general characteristics of the latter type of cooling tower under various conditions of operation, rather than to represent the results of a highly scientific test.

DISCUSSION

Geo. J. Foran. Evidently Mr. Bibbins has intentionally restricted his discussion to the subject of the paper, the cooling tower. He has, however, presented certain tables which, without discussion, are liable to be misleading with reference to the condensers and general cooling-tower condensing situation.

2 The paper discusses the tower quite fully, but classifies the condenser as good, bad or worse without discussion. This is made possible by assuming that the various condenser results obtained are simply a question of condenser design. This permits the inference to be drawn that the various results can be obtained at the same, or practically the same, cost, which is incorrect. Some of the results stated are possible of attainment, but would not show profitable investment.

3 It is impossible to differentiate the tower and condenser quite so completely as in the paper. Each is strongly influenced by the possible range in operation of the other, and I would like to show just how the relative sizes and consequent costs of the plants will be modified by the results desired.

4 Observers agree that the heat transferred through condensing surface varies directly with the mean temperature difference between the two sides of the tubes. Whether this mean should be arithmetical or geometrical is immaterial for the present discussion, and for simplicity I have selected the arithmetical mean.

5 It is unnecessary to assume condensers of varying grades of design and efficiency; in fact, it hopelessly complicates the question, and for my discussion I have assumed a condenser of uniform design and maximum efficiency with a varying amount of surface, which will permit us to obtain the various results tabulated by Mr. Bibbins.

6 The fairly universal practice for high-vacuum work for the past few years has been that for a 15-deg. rise in temperature of the incoming circulating water, during its passage through the condenser, it will be brought to within 15 deg. of the temperature corresponding to the vacuum. The proposition is frequently made to add only 10 deg. to the water and bring it to within 10 deg. of the vacuum. This is perfectly feasible, but we must see what this involves.

7 It means, first, that if we must carry away the heat from the steam by increasing the temperature of the circulating water 10 deg. instead of 15 deg., we must have 50 per cent more water with consequently larger and more expensive circulating plant and piping. With a 15-deg. rise to within 15 deg. of the vacuum temperature, the mean temperature difference between the steam and water side of the tubes will be 22½ deg. With a 10-deg. rise to within 10 deg. of the vacuum temperature, the difference will be only 15 deg. or, in the latter case 50 per cent more surface will be required.

8 Following the 28-in. vacuum line in Fig. 7, it will be noted that Mr. Bibbins has added practically 15 deg. to the condensing water and has given three curves—one for a good condenser with a temperature difference of 10 deg.; a very efficient condenser, 5 deg.; a perfect condeser, 0 deg.

9 Let us consider only the perfect or maximum-effect condenser with varying surface to produce the results named. For the zero-degree curve the mean difference between the steam and water side of the tubes will be 7½ deg; for the 5-deg. curve this becomes 12½ deg. and for the 10-deg. curve, 17½ deg. Or, if we should take the case

where we add but 10 deg. to the water, these three mean differences would become 5 deg., 10 deg., and 15 deg. respectively, so that the condenser for the zero-degree curve would have twice the surface required by the condenser on the 5-deg. curve and three times the surface required for the 10-deg. curve.

10 While there are several plants which report a circulating delivery temperature at approximately the temperature of the vacuum, it is evident that no plant should depend upon such a performance to obtain the economical results upon which the plant investment is based, as this would require absolutely perfect test conditions in every day operation; it would give no leeway at all and would result in too wide a variation in performance for a slight falling off in operating efficiency. Even a slight air leak would result in lowering the temperature in the vacuum space 5 deg., with a consequent loss in heat head and reduction in heat transference, owing to the presence of the air itself. These matters must be considered in addition to the question of cost.

11 Again, following the 28-in. vacuum line in Fig. 7 until it intercepts the 10-deg. curve, it will be found that it calls for water at 75 deg., the 5-deg. curve calls for 80 deg. and the zero curve for 85 deg. All these conditions assume that these results depend only upon the condenser, and if I understand the table correctly, call for the same quantity of steam and water, the temperature of the circulating water, it will be noted, being raised 15 deg. in each case. The author also assumes that the water is cooled to the temperature of the outside air.

12 Although I am sure that the author does not intend to convey the apparent meaning, the further statement is made that this calls for a fixed cooling tower performance; in other words, as I understand it, that the size of tower and the performance will be the same, to cool a given quantity of water through the same range in temperature, irrespective of the temperature of the air.

13 Let us follow this a little further, and in line with the general assumptions, assume for this purpose that the hot air leaves the tower at the temperature of the hot water and 100 per cent saturation. By reference to psychrometric tables it will be seen that each cubic foot of air at 70 deg. temperature and 70 per cent humidity, when increased to 85 deg. and 100 per cent, will take on 7.15 gr. of moisture, whereas a cubic foot increased from 47 deg. and 70 per cent to 62 deg. and 100 per cent, will take on only 3.575 gr. of moisture; that is, although the temperature is increased 15 deg. just the same, the air carries away

but one-half the moisture at the lower temperature, showing that twice the air capacity of tower efficiency will be required at the lower temperature. This is better understood when we consider that within the usual air temperature ranges, the moisture-carrying capacity of the air is doubled for each 22-deg. rise in temperature. To be brief and to avoid confusion, I have used the ordinary nomenclature, which is scientifically incorrect. We all understand that it is the space and not the air which is saturated, but this splitting of hairs would not affect the point under discussion.

14 I have purposely neglected the several minor considerations as they affect the question to a very small extent. For example, the volume of the air entering the tower at 70 deg. and 70 per cent humidity, and leaving at 85 deg. and 100 per cent humidity, is increased nearly 5½ per cent, due partly to the increased temperature and partly to the reduced pressure of the air itself, owing to the increased saturation and vapor present. It is well known that the cooling tower performs its work *principally* by the withdrawal of heat from the main body of water which provides the latent heat for the evaporation of a small portion of the water carried away in the form of vapor as increased humidity of the cooling air.

15 Temporarily omitting the *perfect* plant, let us consider an *average* operating plant in a location having air at 70 deg. and 70 per cent humidity. The usual cooling-tower turbine plant would carry a vacuum of 27 in. with water cooled from 100 deg. to 85 deg. If it is desired to cool this water from 90 deg. to 75 deg., this would permit of carrying a vacuum of 27¾ in. with the same amount of surface and water, but would require an increase in the quantity of air and of tower capacity of approximately 50 per cent. If it is desired to cool the water through only 10 deg. that is, from 85 deg. to 75 deg. and to bring the water within 10 deg. of the vacuum (28¼ in.) this would call for 50 per cent more water, 50 per cent more surface and over 100 per cent more air and cooling tower capacity than for the usual 27-in. vacuum plant.

16 There are hardly two plants which have quite the same determining factors. The determination as to the advisable vacuum and plant must be decided in each case, but there are few plants where the conditions would warrant the installation of a plant to produce the maximum vacuum under the most severe conditions.

17 With reference to the type of tower with fans in the stack, as shown in Fig. 2, the Worthington Company installed their first tower of this type with rope fan drive, in 1900, and recent reports indicate as

good results as when the tower was installed. As a general proposition, however, there are several questions to be considered in comparing this type. There is a saving in the number of fans over the arrangement with the fans below the tower filling, but the fan operates in the hot, highly saturated air, is more or less inaccessible and out of sight, and therefore will not receive the best of attention. It requires good installation and is more difficult to maintain in good condition owing to the fact that it is an exhauster. Any of us would prefer to install a pressure fan rather than an exhauster; the capacity of the fan in the stack must be somewhat larger for the reason that as neither the circulation or the surface efficiency is improved, the total volume of free air required is the same, this being handled at a less pressure and higher temperature and humidity.

18 Comparing the fan and natural-draft towers, there are few, if any, locations where high results are desired, where the natural-draft tower could be selected. A little calculating will convince any engineer that the draft is principally due to the wind velocity over the tower. Study of the meteorological tables will show that in most power centres, except in very few locations, the wind velocity is much greater in winter than in summer—just the opposite of our requirements. This is clearly demonstrated in the operation of any fan tower from the fan speeds permissible at different seasons. It must be remembered that with a tower of the same height the wind assistance is the same for either type of tower. There are many locations where a so-called combined tower can be used if the additional expense is warranted, but strictly speaking, the operation cannot be combined. It must be used either as a natural-draft tower or as a fan tower, but if the fan is operated at all, all the air must pass through it, whether the fan is located above or below the filling.

19 I do not see how there can be any induction in the tower shown in Fig. 14. The object of the tower is to get sufficient pressure below the filling to force through the requisite amount of air, but this pressure must be uniform in the entire space below the filling in order to obtain complete surface efficiency, and under such conditions air would leave rather than enter the tower through any additional openings to the outside air.

20 The Worthington Company make a so-called combined tower which permits of two water levels in the cold well. At the lower level the air enters through the fan at rest and below the lower plates of the tower shell above the water. At the higher level the lower plates are sealed and all the air enters through the fans, which can be operated

at the speed necessary to supply the additional pressure required by the low wind draft. This is also accomplished by the use of additional draft doors.

Prof. William D. Ennis. Will Mr. Bibbins explain in more detail the derivation of the curves in Fig. 7? The tower must provide cooling sufficient to absorb the heat liberated with the exhaust steam, viz., 939 B.t.u. per pound. The amount of cooling in each case would then be 939 divided by the weight of circulating water per pound of steam. On this basis, the maximum temperatures of entrant air agree closely with the curves at 27-in. and 28-in. vacuum, but are about 1 deg. higher than the curves indicate at 29 in., and 2 deg. or 3 deg. higher at 26 in. The curves should apparently be more nearly straight.

2 The paper gives unusually complete and valuable data on many phases of cooling tower operation, but it is to be regretted that the matter of loss of water has not been dealt with in more detail. This is perhaps the most vital question. Manufacturers are sometimes asked to guarantee a limit of loss, but it would be just as logical to ask for a guarantee as to the value of π. A rough estimate often offered is that the loss will not exceed the amount of boiler feed water.

3 Mr. Bibbins gives data from three plants: that at Duquesne, in which the makeup water was from 10 to 20 per cent; the Potosina plant, in which the loss of vapor by windage was occasionally as much as 10 per cent of the volume (of water?) passing through the tower; and the Detroit natural-draft plant, in which the vaporization loss was 2 per cent of the water passing through; practically equal to the weight of boiler feed. The average cooling per hour was $(293{,}530 + 5910.6) \times 16.23 = 4{,}860{,}018$ B.t.u. Each pound of water vaporized, if we neglect the cooling effect of the air, must then have absorbed $\frac{4{,}860{,}018}{5970} = 816$ B.t.u. This is the nearest to a reasonable result I have ever seen in a cooling-tower test.

4 Usually, and this apparently applies to the two other cases cited by Mr. Bibbins, the loss of water is far greater than theory indicates as necessary. The cooling of the water is accomplished by (*a*) the absorption of heat by the air and (*b*) the evaporation of a portion of the water. When the minimum temperature of the air equals or exceeds the maximum temperature of the water, the first effect becomes zero. When the air is initially saturated, the second effect becomes zero, except as the air is heated during its passage. Under

the limiting condition at which there is no direct transfer of heat to the air, the necessary volume of air is increased, and the loss of water does all of the cooling; but the proportion lost need not exceed, in theory, the quotient of the range of cooling by the heat of vaporization, and the use of screens unables us even to reclaim some of the otherwise lost vapor. Why is it that almost invariably the make-up water greatly exceeds the amount thus computed as necessary? It is inferred from Par. 34 that Mr. Bibbins has considered this question of cooling by evaporation, in which case some exposition would be desirable.

Henry E. Longwell. Very early in 1884, under the direction of John C. Dean, of Dean Brothers Steam Pump Works, I made drawings for a cooler that was built for the Kane Milling Co., Kane, Ill. I am told that it was the first one erected in the United States, and it is, at any rate, a well-authenticated case of a very early installation. The plant was operated for only two years, being then destroyed by fire, but so far as I can remember the installation performed in a very creditable manner, especially considering the primitive state of the art at that time.

2 There are probably many engineers who will take issue with the author if he means that the cooling tower field is yet comparatively unexplored. For ten years or more the cooling tower has been on a strictly scientific basis. Its design and construction constitute a branch of engineering that is just as distinct and as well developed as any of those which deal with other specialties such as gas engines, steam turbines and the like. When we consider that one builder alone has constructed about 2000 cooling towers which in the aggregate are capable of cooling condensing water for about 3,000,000 horsepower, we must admit that this device has progressed a long way beyond the rudimentary stage.

3 It is not excessive cost or lack of knowledge that has restricted the use of cooling towers in the United States. It is because nature has been so good to us that the conditions in which cooling towers are desirable or necessary are comparatively rarer than in the less favored and more congested European countries, where these devices have reached the highest state of development.

4 I regret that the author has not presented in exactly the same form the two tests of the cooling tower described. In Table 4 is given a complete log of the principal observations made at approximately hourly intervals; in Table 5 we have only the average of all the

observations made over a period of 24 hours. The two tests were made under such widely different conditions that they afford no proof as to whether the performance of the tower was any better or even as good with its full complement of cooling surface, as it was with only three-fifths of it. During the test with only three-fifths of the cooling surface installed, the average load was nearly 80 per cent greater, and the average quantity of water circulated per hour was nearly 35 per cent greater than on the test with all of the surface installed.

5 Referring to Fig. 11, the indications are that the added cooling surface served no useful purpose. Indeed if the diagram means anything at all, it means that for the same temperature head the product of the heat dissipated per square foot of surface per hour multiplied by the proportion of the cooling surface installed, is practically a constant; also, that for equal temperature heads, the number of degrees cooling is practically the same.

6 In Fig. 12 in which temperature head is plotted against degrees of cooling, the lines corresponding to 3/5 surface and 5/5 surface, coincide so nearly that one could hardly say that they depart from each other by more than the limit of the normal error of observation.

7 Fig. 13 at first sight seems to indicate that at hot-well temperatures below 120 deg. the cooling was considerably greater with 5/5 than with only 3/5. But we know that on the test with only 3/5 of the surface, the amount of water circulated was very much greater than with 5/5 surface. Comparisons of this sort are misleading unless the quantity of water circulated per hour and the temperature of the incoming air are the same in both cases.

8 The inconsistency of the curves in Fig. 13 will become apparent if we extend the straight line curve for 3/5 surface until it cuts the line of zero cooling. This will indicate that at a hot-well temperature of a little above 85 deg. the water would not be cooled at all, although we know from Table 4 that the temperature of the incoming air was at no time higher than 35 deg. The inference would be that water entering the tower at a temperature below 85 degrees would be warmed by coming in contact with air at or near the temperature at which water freezes.

9 The indications are that the tower is too small for the work, and that its performance is limited, not by the amount of cooling surface, but by the weight of air that can pass through it in a given time. After all, it is the air that carries off the heat, and the quantity of air passing through the tower is just as important a factor as is the area of the cooling surface.

10 The tower described occupies 200 square feet of floor space, and is rated at 900 h.p. Assuming 15 lb. of steam per h.p. hour, the tower would have to cool sufficient water to condense 13.500 lb. of steam hourly. A natural draft tower designed by one of the most experienced builders of this class of apparatus, would for this same duty occupy a space about 29 by 24 ft., or nearly $3\frac{1}{2}$ times as great as that occupied by the towers described. It would also be from 7 to 10 ft. higher, which would give a more powerful draft.

11 Referring again to Fig. 12, it will be seen that the temperature of the water leaving the natural-draft tower is from 40 to 70 deg. above that of the incoming air. On this same diagram are curves which purport to show the performance of the forced-draft tower briefly referred to in Table 3. It would appear from these curves that the forced-draft tower under favorable weather conditions cools the water to within 3 or 4 deg. of the atmospheric temperature. Under unfavorable weather conditions it appears to cool the water to within 15 to 35 deg. of the temperature of the atmosphere.

12 The cost of the forced-draft tower is given as $2.60 per h.p. as against $1.50 for the natural-draft tower. However, if the comparative results as shown in the diagram Fig. 12 are dependable, it would appear that the forced-draft tower was well worth the additional cost, and a little bit more.

13 In Fig. 7, the author purports to show the maximum temperature of inlet air permissible for various vacuums. This diagram really shows the maximum temperature of cooling water to produce a given vacuum on the assumption that we limit the number of pounds of cooling water per pound of steam condensed, to the arbitrary figures set down in the lower right-hand corner of the diagram. The temperature of the atmosphere is not necessarily the limiting temperature to which the water may be cooled. It is well known that with low humidities, cooling towers may reduce the temperature of the water to several degrees below that of the atmosphere. And there is no law of nature that stipulates that we may circulate no more or less than 100 lb. of condensing water per pound of steam to produce a 28 in. vacuum, or 60, 40 and 30 lb. per pound of steam to produce respectively vacuums of 27, 26, and 25 in.

14 Fig. 15 is doubtless interesting, but as no reference to its usefulness appears in the paper it is difficult to see wherein it is pertinent.

15 I would point out that the diagram Fig. 1 shows that on two days in June 1906 the *average* temperature exceeded 90 deg. According to Table 1 on the following page there was not a single day during

that month on which the *maximum* temperature reached 90 deg., to say nothing of the average. If there were 10 days in the month of June 1906 on which the temperature exceeded 75 deg., it is difficult to see why there must not have been at least as many days on which it exceeded 70 deg. The quantities set down in the columns headed "Average for Month" require some explanation to make them intelligible.

16 The theory of cooling towers is simple, and any one who has a reasonable acquaintance with that branch of natural science which deals with heat, may easily know it a little or even very well. As far as the theory itself is concerned it would be hard to improve on the clear, concise and generally masterly presentation of the subject by F. J. Weiss, inventor of the well-known Weiss condenser, which appeared in a book entitled "Kondensation," published in Germany about ten years ago. But as in all branches of engineering, the coefficients by which theory is reduced to practice, are the property of the few, who by special application and practical experience have come to know the subject profoundly.

Barton H. Coffey.[1] The advent of the turbine with the high cost of fuel in steam plants and the increasing cost of water for cooling purposes in urban installations of refrigeratin apparatus, are making the cooling tower a necessary means of economy.

2 As the author remarks, the literature upon the subject is scanty; in fact, with the exception of C. O. Schmitt's paper before the South African Association of Engineers in 1907, there is scarcely anything extant that I know of, worthy of the name.

3 I do not wholly agree with Mr. Bibbins' presentation of the meteorological conditions to be met by cooling towers, as given in Fig. 1 and Table 1. The comparison of average humidity and temperature, as given by the weather bureau, is a little misleading, as the humidity observations are made at 8 a. m. and 8 p.m. only. In lieu of hourly humidity measurements, I think it better to take the average aqueous pressure at 8 a.m. and 8 p.m., as it is known that this quantity changes slowly, and from this the hourly humidities can be calculated. It will then be found, of course, that as the temperatures advance toward midday, the humidity falls, thus tending to maintain average thermal conditions with respect to cooling towers and explaining the approximately uniform results actually obtained. The mean aqueous pressure for July, covering a number of years, work out about as follows:

[1]With Edwin Burhorn, 71 Wall Street, New York.

TABLE 1 MEAN AQUEOUS PRESSURE

	Actual Aqueous Pressure
Boston, Mass	0.542 in. mercury
Philadelphia, Pa	0.614 " "
Salt Lake City, Utah	0.296 " "
St. Louis, Mo	0.648 " "

At St. Louis, therefore, where the mean maximum temperature for July is 88 deg., the relative humidity would be 49 per cent against a mean humidity of 66.1 per cent, as given by the tables, which is distinctly a more favorable condition for cooling towers.

4 While on meteorology, I would like to call attention to the statement in Par. 15*b*, that the tray or atmospheric type of tower cools only by means of "transverse air currents from the side", the obvious deduction being, that without wind this type of tower fails. As a fact, in a dead calm the efficiency of all forms of tower falls off, but this condition is of small practical account, as in the interior region the percentage of calm rarely exceeds 2 per cent of the month and on the seaboard is practically unknown. However, in a dead calm the towers still continue to work, due to an ascending column of warm air and aqueous vapor over the tower and a corresponding horizontal inflow of cool dry air. This condition must exist, otherwise the entire space surrounding any tower would become filled with warm saturated air and all cooling would cease. In a forced-draft tower for example, the fan would be simply circulating air having no capacity for absorbing heat. Apropos of this, I have records from an atmospheric tower on refrigerating work for the entire month of September 1907, taken with recording thermometers, in which the cooling water from the tower was maintained at an average of 75 deg., never exceeding 80 deg., with a cooling range of about 10 deg.

5 In Par. 13*b*, among the elements of design, Mr. Bibbins advises "Avoid free falling water. It should be distrubuted so as to descend clinging to some form of wetted surface." I would like to know the basis for this statement, as probably by far the largest number of towers in use throughout the world employ the principle of finely divided falling water, as, for instance, the various forms of atmospheric and chimney towers in Europe, South Africa and this country.

6 As 75 to 85 per cent of the cooling is due to evaporation, which can take place only at the surface in contact with the air, the form of cooling surface is of great importance. In a cooling tower with free-falling water, the cooling surfaces consist of the hurdles or decks and the exposed surface of the falling water. Experiments show the

weight of a drop of water to be about three-fourths of a grain, the diameter of the corresponding sphere being 0.178 in. A gallon of water properly distributed will therefore expose about 54 sq. ft. of surface. If we know the flow per second and the time of fall in seconds, properly corrected for atmospheric retardation, we can calculate the exposed surface in the water, which, added to the fixed wetted surface, gives the total cooling surface in the tower. The efficiency of the surface in the falling water is greater than the fixed surface, due to the greater velocity of the air relative to the water surface, due to the motion of the drops.

7 The question of type of surface, in my opinion, is one of expediency to be determined by the conditions of operation. Fixed surface is undoubtedly more compact and when skillfully designed opposes less resistance to air currents. On the other hand, it involves weight, greater difficulties in distribution, and where oil is present in cooling water, it becomes coated, the capilarity is destroyed and the water film is reduced to streams, thus greatly lessening the water surface exposed.

8 If the atmospheric form of tower is to be employed, it is hard to conceive of any form of surface, save drops, that would be exposed to the wind from any direction; and where space is available for sufficient surface, the temperature reduction called for can always be attained.

9 In a test by the speaker of an atmospheric tower circulating 440 gal. per min., with air at 93 deg. and humidity 34 per cent, the water was cooled from 80 deg. to 74 deg., or within 3 deg. of the wet bulb, which is the limit of atmospheric cooling.

10 With reference to Mr. Bibbins' remarks on the effect of temperature range on the size of the tower, I beg to submit a few figures on the volume of air required at 80 deg. and 80 per cent humidity to absorb 1,000 B.t.u. when the air can be heated to the following final temperatures and saturated:

TABLE 2

Class of Work	Final Temp. Air	Cubic Feet
Refrigeration	88 deg	985
Steam Condensing 27 in. vac.	100 "	429
Steam Condensing 26 in. vac.	110 "	267

This shows the enormously increased quantities of air required as the lower ranges of cooling are approached, and also shows the particular advantage of the atmospheric tower for refrigeration work, in saving the power necessary to handle this large volume of air. As an example:

11 With air at 80 deg. and 80 per cent humidity, to cool 600 gal. of water per min. to 80 deg. would require about 70,000 cu. ft. of air per min., requiring about 17 brake horsepower in a fan tower. An atmospheric tower of like capacity, having 960 sq. ft. wind exposure, would receive 248,000 cu. ft. of air per min. at a velocity of 4 miles per hour. In steam condensing, with a limited space the forced-draft tower is, of course, the only available type.

Carl George de Laval. The author states that the present high prices constitute the greatest obstacle to the use of cooling towers, and, further appears to give the impression that the cooling tower is a makeshift and not a permanent apparatus.

2 There are three classes of towers, forced-draft, natural-draft and a combination of both, the last-named being used either way depending on the season of the year. The selection of the type should depend on climatic conditions, cost, etc., a dry climate being best suited for a cooling tower.

3 The author states that the costs range from \$4.80 to \$6.93 per kw., which appear to be slightly higher than market prices, the reason perhaps being that the author had imposed severe conditions when asking for bids on cooling towers, thereby increasing the costs.

4 Let us assume a plant of 1,000-kw., consuming 19,000 lb. of steam per hr., basing the condenser performance upon the ordinary 10-deg. difference in a counter-current jet condenser, and upon a 27 in. vacuum, with air at 70 deg., and 70 per cent relative humidity. A cooling tower with interlocking pipe filling can be built approximately 19 ft. by 35 ft., fitted complete with fans, for about \$5 per kw., and a wood-filled tower about 21 ft. by 35 ft, for about \$4.50 per kw.

5 The author is correct in stating that installations are not being sufficiently studied, and this, no doubt, is the principal cause for the failure of cooling towers and has prevented their more general adoption. It is not sufficient merely to obtain information as to maximum load, steam consumption, maximum temperature and humidity, but it is necessary to know whether these maximum load conditions must be met at the conditions of maximum temperatures and humidity, and if so, for how long a time.

6 Let us assume that bids are asked for a cooling tower for 8,000 kw., the conditions named being an air temperature of 75 deg. and 75 per cent humidity, 27 in. vacuum, no time being stated when this load of 8000 kw. is likely to occur, and what its duration is. The real facts may be that this load comes in winter only, and that in

summer probably not over 5000 kw. would be required during the evenings, while the summer mid-day load might not be over 20C0 or 3000 kw. Under such conditions a tower calculated for a 5000-kw. summer load would be ample for an 8000-kw. winter load, and if the installation was made on the basis of 8000 kw. the year round, the cooling tower would be too large and expensive, and the cost per kilowatt of maximum load would be too great.

7 The maximum mid-day temperature and humidities likewise should not be the basis of consideration with maximum loads, as the electric lighting plant maximum during summer should instead be based upon 8 p.m. temperature and humidities. One sometimes sees the requirement to handle maximum loads at an atmospheric condition of 90 deg. and 80 per cent relative humidity—a condition that may never be reached in the particular locality where the tower is to be installed.

8 Most of the towers described by the author appear to be home-made or makeshift towers, for instance, the tower shown in Fig. 6, and, installed at Butte, Mont., having a cross-board filling and a wooden stack for natural draft. The design is such that it will lose considerable of its efficiency as it continues in service, and the boards, as well as the upper stack, will warp, admitting cold air above the filling and tending to kill the draft upon which such a tower depends for its efficiency. The warping of boards will also cause leakage through the sides of the tower, the leakage being carried by any strong breeze, and thrown against surrounding buildings and territory, where during winter it may freeze into a heavy mass.

9 Referring to preceding discussion on the design of towers for maximum atmospheric conditions, one will note in the temperature ranges in Table 2, for the Butte tower, that the atmosphere was over 80 deg. during less than 3 per cent of the total time of the year, so that such conditions can hardly be used as a basis for calculation. Atmospheric conditions at Pittsburg during the four months from May 15 to September 15 average approximately 70 deg, and 70 per cent, which appears to be about a standard basis for cooling towers.

10 Par. 9 refers to the use of cooling towers for handling jacket water of gas engines, the temperatures being about the same as those encountered in ice plants, and higher than in the case of steam condensation. Several installations show this temperature to be from 156 deg. to 111 deg., and 130 deg. to 80 deg.

11 Par. 10 and Par. 11 state that the difference between the theoretical steam temperature and the temperature of the circulating

water varies from 10 deg. to 50 deg. The usual jet condensers and surface condensers give about 15 deg., and cooling towers for reciprocating engines are usually based on a 24-in. vacuum, with circulating water cooled from 125 deg. to 100 deg. and an air temperature of 70 deg. and 70 per cent relative humidity. Counter-current condensers give about 10 deg. difference, the circulating water being handled under the same conditions of vacuum, with a temperature range from 130 deg, to 105 deg., instead of from 125 deg. to 100 deg., which of course gives an easier condition for the cooling tower.

12 It is a well-known fact that an efficient condenser must be installed in order to get good work from a cooling tower, it being an advantage to the tower to have the temperature of the hot water and the cold water as high as possible. For instance, taking examples of the two conditions, both 1000-kw plants consuming 19,000 lb. of steam per hr., at 24-in. vacuum, and an air temperature of 70 deg. and 70 per cent relative humidity, one plant being based on a 40-deg. difference between the exhaust steam and the outlet circulating water, which requires the water to be cooled from 100 deg. to 75 deg.; the other plant being based on a 10-deg. difference between the steam and the water, the water being cooled from 130 deg. to 105 deg. In the former case, for the same load, vacuum and air temperature, we require an interlocking pipe-filled tower, 22½ ft. in diameter by 35 ft. high, having four 96 in. fans; whereas in the latter case with only a 10-deg. difference we can do the same work with the tower 13 ft., 6 in. in diameter by 35 ft. high, having one 120 in. fan. The efficiency of the condenser therefore makes a very decided difference in the size of cooling tower.

13 Under *c* in Par. 13, the author apparently refers principally to towers with wood sides, having a wood structure within the outside boarding. It is very important that the filling must come close to the side of the tower. Particular care should be taken in erecting towers to see that pipes are first laid around the outside edge as closely to it as possible; otherwise, there will be a short circuit of cold air around the side of the tower and a loss of efficiency. This condition, while bad enough in the forced-draft tower, is much worse in towers of natural-draft type, because this air will seriously reduce the draft by mixing with and cooling above the filling the heated air upon which the draft depends.

14 As to height of working section, it is true, as the author states, that the height is important, and the distance of the elevation of the water should be kept as low as possible. A pipe-filled tower is

13 ft. 4 in. deep with a drop at the bottom of from 6 ft. to 11 ft; according to the size of the tower. With a distributor operating head of 5 ft., this gives the largest towers a maximum pumping head, plus friction in the piping, of 29 ft. 4 in. against approximately 38 ft. as required in the experimental natural-draft tower at Detroit, shown in Fig. 9. The horsepower necessary to pump the water the additional 8 ft. 8 in. in height will offset the usual fan horsepowers, making a natural draft tower of this type more expensive to operate than a fan-draft tower.

15 As to the mat of wood swelling and being thrown out of place, I would state that towers have been built with a cross-board wood filling, and four of these have been in satisfactory operation since 1904. In these towers were used 2 in. by 2 in. verticals at intervals through the filling, with the boards nailed so as to hold the filling in place and prevent distortion or formation of large open gaps through warping of the filling.

16 The cooling towers illustrated in Fig. 3, are furnished with perforated pans and have free-falling water, the sides being screened. This tower depends for its efficiency upon a cross breeze and is very inefficient in still air as the air cannot rise within the tower on account of the pans. A strong breeze will blow most of the water out through the sides of the cooling tower, in spite of the screen. The tower shown in Fig. 4 occupies considerable space, and also requires additional space in the immediate vicinity because of loss of water through windage. The tower illustrated in Fig. 5 is evidently much less efficient than that in Fig. 4, because of the large amount of free-falling water. The free-falling or splashing of water is a very inefficient method of cooling. Water for proper cooling should always be brought down in contact with a surface so that it will descend slowly and thus have close and intimate contact with surrounding air.

17 In Par. 26, the author gives the total cost of the Detroit tower, erected complete and including filling, distributing pipes, foundations, etc., at \$1350. It appears that the steel work, if made of at least No. 10 gauge, would weigh approximately 20,000 lb., which at 6½ cents per pound, which is about as low a rate as mentioned, would require an expenditure of \$1300 for steel work alone. The lath filling and the work of assembling and installing this tower, would cost about \$400; the timber supports, distributing piping, etc., about \$250; concrete foundations an additional \$250, or a total cost of \$2200. Assuming a load of 1000 hp. with 19,000 lb. of steam per hour, vacuum 24 in., with a temperature difference of 10 deg., the

circulating water being cooled from 130 deg: to 105 deg., air at 70 deg. and 70 per cent humidity, a pipe-filled cooling tower of the fan type, measuring 13 ft. 6 in. by 35 ft. could be installed for about $2500.

18 The results of the test given in Table 5, with atmospheric temperatures of from 18½ deg. to 30 deg., are not complete for a natural-draft tower, as such towers fall off in efficiency very rapidly when the air temperature is raised. The results at temperatures from 70 to 80 deg., would not be so favorable.

19 In Par. 30, the condition of scale covering the wooden filling would be experienced in any tower, and is usually encountered where well water is used to make up in cooling towers for refrigerating plants. The scale forms a protecting coating in a pipe tower and prevents possible rusting of the pipe filling.

20 In Par. 32, the author refers to possible advantages of a slotted pipe as compared with spouts on a distributor arm, in regard to clogging. The spouts used by some first-class designers are 1 in. in diameter and are consequently much less liable to clog than are pipes having a ⅛ in. slot in the top.

21 In Par. 15, the author refers to the use of sprays over a pond. This seems a very simple apparatus, but it must be realized that the sprays require from 15 to 20 lb. pressure at the nozzle and so consume more power than required for circulation through a cooling tower of the fan type, and in most cases as much power as is required both for the circulating of the water and for the driving of the fan.

22 The arrangement of cascade or cooling sprays on a roof as described by the author is not new. The installation was in use by J. H. Stut of San Francisco, previous to 1892, being placed upon the roof of a factory. Galvanized troughs, 5 ft. wide were arranged in tiers on a slight incline so that the water traveled back and forth a distance of about 2,000 ft. before being returned to the condenser. An arrangement of falling from one trough to another, these troughs being spread out upon a roof was used at the old Budweiser Brewery in Brooklyn previous to 1890. The sprays and roof troughs are alike open to the objection that if there is a strong breeze the water is carried all over the surrounding neighborhood and if there is no breeze, a heavy fog quickly collects at the point of spray and thus greatly reduces the amount of cooling.

23 Referring to various types of filling illustrated in Figs. 8*a* to 8*c*, Fig. 8*a* offers too serious an obstruction to the draft within the tower, closing more than 40 per cent of the space necessary for verti-

cal circulation of air, as against 3 per cent covered by interlocking pipe-filling or 25 per cent by wood filling. The cascades as illustrated must fall as shown in the sketch in order to operate efficiently, that is, the water must strike the pans on the next lower section of the filling, but this they will do only if the amount of water supplied is practically constant, otherwise it is liable to spill over several rows of filling, and result in quick descent and consequent loss of efficiency.

24 The filling illustrated in Fig. 8*b* is that used by Henry W. Bulkley, and depends upon a cross steam of air, as in the tower shown in Fig. 3. It is open to exactly the same objection as the latter tower.

25 The filling 8*c* will cause large quantities of free-falling water between the several courses and will result in inefficient operation. The filling 8*e*′, a wooden cross-board type, is apparently good. It requires additional expense in placing, but evidently will save something in fan horsepower. The filling 8*d* offers a bad obstruction to the draft on account of deflecting the air alternately to the right and left. The water also will evidently flow down the top side of the board, whereas the air impinges most strongly against the lower side of the board.

26 The filling 8*b* is the same as 8*b*′ and is good. The filling 8*g* is open to the objection of having no redistribution,—the water distributed at the top, however unequal it may be, must remain unequal from top to bottom. The filling 8*f* has one redistribution at the center. Otherwise it is open to the same objection as 8*g*.

27 In Par. 23, and also in the footnote, it is stated that ball bearings are difficult to keep in good condition. Ball bearings are not used in modern towers, a floating water-step bearing being used instead.

28 Referring to Par. 41 and Par. 42, a combination type tower may run with natural draft about eight months during the year. At a plant in Newark, N. J., a combination type tower with side doors operates on an 800-kw. load nine months of the year with natural draft, and requires 25 hp. during the remaining three months of the year.

29 As to Par. 46, efficient condensers are more badly needed than efficient cooling towers. Cooling towers have reached as high an efficiency as can be expected, but most plants now operating with direct jet condensers delivering into the towers, could obtain much higher vacuum or handle greater loads at the present vacuum, if condensers of the counter-current jet type, or the more efficient baffled-surface condensers were substituted in place of the condensers originally furnished.

30 As to Par. 48, the temperatures are practically the same as in ice plants. In order to get the temperature head mentioned, it is more economical to circulate the water for the ice plant first through the ammonia condenser and then through the steam condenser delivering to the cooling tower, than to have two towers handling separately the water of the ammonia condenser and that of the steam condenser.

31 The open wooden towers referred to in Par. 50 are not restricted to points of low humidity; but as already mentioned, they require much open ground, not only on account of their size, but also for wind effect, and that surrounding buildings may not be drenched with water blown from the tower.

32 As to Par. 51, the tower best adapted to natural-draft work is the one which offers the least resistance to the ascending current of air. In Par. 52, no temperatures are given to substantiate the statement of heat dissipation by lath mat construction.

33 As to Par. 53, one cannot endorse the fan booster or induction type when combination towers can be made that will give better results and that are surely preferable for overload conditions.

34 The largest number of towers in this country are of the forced-draft type, while European practice tends towards natural-draft towers. It is thus apparent that there can be no standard of type or size, because of difference in climates; each installation must be considered as a separate problem.

E. D. Dreyfus. In Par. 10, Mr. Bibbins says, "But in practice from 10 to 15 deg. difference exists, depending upon the type of condenser and the volumetric ratio of water to steam." I wish to supplement this by adding that it does not depend altogether on the volumetric ratio. Another important factor is the effectiveness of air removal. Those who have had considerable experience in condenser work find that the more effectively the air is withdrawn, the nearer the theoretical vacuum is reached. By this means it is possible to operate with a diminished volumetric ratio as the temperature rise is increased.

2 Exception is to be taken to Mr. Foran's remark that perfect condenser operation entails much greater experience, which might be implied as generally applicable. This is true only of surface condensers. In cooling tower practice, the conditions are extremely favorable to the use of the more simple jet type. The more efficiently this latter type is operated, that is, the nearer the discharge water

is brought to the temperature of the exhaust steam, the smaller is the volume of water necessary, since volume and temperature rise are component factors of the B.t.u. extraction. Therefore, with less volume of water handled, the size of the condenser may be reduced and consequently furnished at a smaller cost.

3 A remark made by the author in presenting the paper, that an inefficient condenser and an efficient cooling tower go hand in hand, bears further explanation, although the statement was modified somewhat subsequently. With an inefficient condenser, the vacuum is not likely to be very good, and therefore, with the higher temperature prevailing in the condenser, the water might pass to the tower at a higher temperature, making it easier for the latter apparatus to perform its work. On the other hand, the statements might be applied with equal, if not greater, force to efficient condensers which are able, for the same condensation, to create higher vacua, besides heating the discharge water up to the same final temperature head as the inefficient type, there being little or no terminal difference in an efficient design at its normal capacity. Moreover, considering the benefit accruing to the prime mover, a smaller volume of water may be used and worked at the same temperature as in the inefficient type of condenser, thus increasing the possibility of the tower. I would qualify the above statement to the extent that it deals with a comparison of condensers designed for the same vacuum, and evidently would not hold for a case where a very poor vacuum was admissible.

4 It might be well to state here that a near approach to the theoretical vacuum is not an impossible condition in actual operation. This implies, of course, that the character of the condenser design—the counter-current type with an efficient air pump—fulfills the requirements. In a test which I conducted last Fall on a 1000-kw. low-pressure turbine equipped with a counter-current jet condenser, the following results were obtained: At three-fourths load with 83 deg. injection water, a vacuum of 28.20 in. (30 in. barometer) was maintained, and the water left the condenser at a temperature of 96.8 deg. The temperature corresponding to the vacuum was 97.6 deg., giving practically one degree terminal difference.

5 I have observed that temperatures of the water leaving the tower were several degrees colder than the atmospheric temperature in warm weather, the difference being as much as ten degrees at times.

6 With the increasing recognition the cooling tower is receiving, it would be desirable to have the Society define a standard basis of

measuring the efficiency of the apparatus. There is a conspicuous lack of harmony of opinions as to what constitutes the governing characteristics of tower performance.

T. C. McBride In the earlier parts of the paper the author would lead us to believe that cooling towers have not received the scientific attention warranted. Reference to the literature on this subject and the work that is being done hardly confirms this fact. A considerable number of manufacturers have for some years past been supplying cooling towers designed on scientific lines, and the proposals submitted by them, particularly on fan-type towers, are intelligently framed and leave no points whatever open to guess work.

2 The paper very properly calls attention to the intimate relationship of condenser efficiency to cooling tower performance, but in doing so is extremely unfair to the condenser—in fact, in speaking of different types of air pump, the author almost leads us to believe that some are so superior to others that the vacuum they create is of a superior kind compared to that created by other air pumps.

3 Condenser engineers now agree that the efficiency of condensers, with regard to the comparison of discharge-water temperature with theoretical vacuum temperature, is as much a question of the average temperature of the vapor in the condenser as its design. The average temperature of the condenser is necessarily determined by the amount of air present therein, and is a direct function of the ratio of the air-removing capacity of the air pump and the volume of air reaching the condenser with the steam. The merit of the air pump cannot therefore be determined either from the vacuum obtained or from the relation of the discharge-water temperature to the theoretical vacuum temperature, but is wholly a question of the capacity of the air pump to handle air at the least expenditure for power, maintenance, interest on first cost and depreciation.

4 It is true that the question of condenser efficiency and air-pump efficiency is somewhat involved with that feature of condenser design having to do with the reduction of air-pump suction temperature, but as all condenser designs should take care of this feature it may be eliminated from the comparison of types of condensers or types of air pumps. It is conceded that the author's division of condensers and air pumps into good, indifferent and bad classes, in accordance with the vacuum and discharge-water temperature obtained, follows lines which have been generally accepted in the past; but a view from an engineering standpoint must consider the

impurities in the steam in the shape of air and non-condensable vapors, before judging any particular type of condenser or air pump.

The Author is exceedingly grateful for the interest shown in the paper and the practical nature of the discussion, which has served to clear up several ambiguities and to extend the subject into channels of inquiry representative of everyday commercial problems.

2 Mr. Ennis deprecates the loss by windage of considerable volumes of circulating water, in excess of that supplied by condensed steam. Theoretically, without windage loss, there should be practically no make-up water required, as an exact thermal balance has been established. But this loss does occur in both forced-draft and open tray type towers, and often to a serious extent. However this is simply a point in favor of the closed natural-draft type of tower, in which the velocities are reasonably low and hence small tendency exists to abstract water from the cycle.

3 The high loss in the Duquesne Lighting Co. Plant, it should be explained, is not due to windage. The hot jacket water can only be partially cooled, consequently enough must be thrown away to lower the temperature by the addition of fresh cold water. The loss at Potosina, however, was entirely due to windage.

4 The curves in Fig. 7 may very possibly be slightly in error, as they were necessarily based upon arbitrary assumptions—hence no attempt was made at absolute accuracy.

5 Mr. Foran evidently has had in mind the surface condenser in discussing possible and probable temperature differentials, whereas the author has referred more particularly to the barometric or jet types, especially in Fig. 7. This should have been stated more clearly in the paper. Generally speaking, it is possible with the jet type to work with much lower differentials than with the surface type. Mr. Foran's deductions regarding the extent of surface required to meet special conditions are therefore entirely proper. This very difficulty which is experienced with surface condensers in meeting the conditions imposed by the best cooling tower practice, only emphasizes in the author's mind the inherent advantages of the jet types.

6 The term "fixed cooling tower performance" could not apply to the construction of the curves in Fig. 7, as it is here used in the sense of efficiency rather than size. The use of "performance" here was in reference to relative cooling effect (deg. fahr.)—not capacity for absorbing heat—for the sake of eliminating another variable in the construction of Fig. 7. The size or capacity for a given con-

dition is simply a function of a heat quantity (B.t.u.) absorbed from the exhaust steam. For a given type of surface and draft velocity, the rate of absorption is fairly constant—a parallel to the constant rate of heat transmission through the tubes, as cited by Mr. Foran.

7 In reference to the Detroit tests, Table V, it should be noted that the condensing plant was not well adapted to the work in view, being an equipment temporarily retained in service from an old plant, too limited in surface and without means of operating air and water pumps individually, as required for economical working. The poor results from this particular plant were therefore distinctly attributable to the temporary nature of the installation, and not to an inherent fault in the type itself, as might be gathered from the reports.

8 In his closing remarks, Mr. Foran seems to confine the use of "natural-draft tower" to the open tray type. It is quite true that this has no application where large capacities or the highest efficiency are necessary. The closed chimney type is not dependent to any extent upon lateral wind velocity, and may be designed to economize space effectively.

9 The point raised by Mr. Dreyfus in regard to the effect of low temperature differentials is well taken. The author's observation that poor vacuum and good cooling go hand in hand applies to a given equipment, but the highly efficient condenser with low differential of course finds the most direct application.

10 The author did not observe or infer that the cooling tower field remains comparatively unexplored, but that certain conditions have tended to render the subject a closed book. This is not the case with engines, turbines, boilers, condensers, etc., so the fact that this condition obtains with cooling towers is not readily justifiable.

11 The two series of tests could not be presented in identical form, as the data were not available in such form. However, the curves, Figs. 11, 12 and 13, were drawn up to facilitate comparison. The first test covered day and peak loads only; the second, the entire 24 hours,—hence a low average load, as Mr. Longwell observes. Because the tower shows a low rate of heat dissipation with the entire surface installed, it should not be inferred that the actual work done was proportionately lower. Considering abscissae (B.t.u.) as equivalent to load (kw.) it must be apparent that for the same load a much higher cooling effect was obtained with the cooling surface complete.

12 For equal temperature heads, the cooling is bound to be the same except when the "lost head" differs, as it does slightly in Fig. 12. This opens up an extremely interesting line of inquiry—a survey

of rates of heat dissipation and humidity in each successive zone of the tower. Which part of the tower does the most work? Assuming air to be discharged exactly saturated at the temperature of exit, what spacing of mats is correct to produce a proper gradation of humidity from, say 70 per cent at entrance to 100 per cent at exit?

13 Regarding the inconsistency of Fig. 13, Mr. Longwell has forgotten to reckon the "lost head" shown in Fig. 12—approximately 40 deg. There is thus a very small discrepancy. However, it is hardly safe to interpolate in such a case. It is already pointed out in the paper that the tower is working at a disadvantage, owing to the extremely poor condenser performance, that imposes an extra burden on the prime mover as well.

14 The circulating water ratios adopted as a basis of the curves in Fig. 7, were so adopted to approximate average practice, otherwise a "family" of curves would replace each single curve shown.

15 Mr. Coffey favors the use of vapor pressure in lieu of relative humidity. The author entirely agrees to this method as more scientific. However, absolute humidity expressed in grams per cubic feet perhaps has a more direct bearing on cooling tower work.

16 The suggestion "to avoid free falling water" should have been amplified in the paper, and Mr. Coffey justly directs attention to it. Compactness or maximum duty for a given size is so essential in restricted locations that the atmospheric type is handicapped, if not debarred, which he himself recognizes in the closing sentence. The paper is directed entirely along these lines of maximum duty, and especially toward the development of the natural-draft type.

17 Mr. de Laval advances the argument that a tower should not have to be designed, rated and purchased *entirely* on a peak load basis. This is entirely in agreement with the author's object in presenting the combined natural-force-draft tower with fan auxiliary for use only during peak loads or during bad weather.

18 The objections of Mr. de Laval to the construction of the Butte tower are, however, not well taken, as the construction is more substantial than as described by him, and several years' service has not developed the defects he mentions.

19 The tests made at Detroit occurred, it is true, during the colder season, but in Par. 29 it is stated that the tower showed very little difference in operation in winter or summer—this on the advice of the chief operator.

20 Tables IV and V present the temperatures asked for to substantiate the assertion of a safe rate of heat dissipation of 200 B.t.u. per square foot per hour for the lath mat construction.

PUMP VALVES AND VALVE AREAS

By A. F. Nagle, Published in the Journal for Mid-October

ABSTRACT OF PAPER

This paper is designed to call the attention of engineers to the need of revising the common notion that "valve-seat area" is synonymous with "velocity of flow." It is evidently the purpose of specifications for pumping engines to secure a low velocity of flow through the valves, thus reducing the head required to force water through the pump; but to accomplish this laudable purpose, special and intelligent attention should be given to the *springs* of the valves, rather than to valve-seat areas. If that be done valve seat areas need not be greater than the plunger area for the vertical triple-expansion pumping engines so largely used in city pumps. A slight economy in both construction and operation could be effected by giving more study to the proper design and strength of springs for pump valves.

DISCUSSION

Charles A. Hague. The practice referred to by the author, of specifying that the area through the pump valves of waterworks engines shall bear a certain relation expressed in percentage of plunger area, is becoming less frequent, and it is to be hoped that it will finally be disregarded altogether. The relation between the plunger areas is merely incidental, because the valve area is a function of the quantity of water to be handled, the important matter being the velocity of the water through the valve seats to fill the plunger chamber as nearly complete as possible under the conditions.

2 The total valve area, or total area of valve-seat opening, ought to depend upon the velocity needed to pass the required quantity of water in a given time. Some authorities advocate a velocity not to exceed 3 ft. per sec., others 4 ft. per sec. and some as low as $2\frac{1}{2}$ ft. per sec. Two factors are to be considered, as follows:

3 First, as to the lift of the valves. The lower the pressure, and the lower the speed of the engine, the higher the valve may lift; on the contrary, the higher the pressure, and the higher the speed of the engine, the less the valve may lift, if a smooth, easy running, economical engine is desired.

4 Second, regarding the circumferential area of the valve space, or the area of the space around the edge of a disc valve, when it is open or off its seat. This is a factor that need not be very seriously considered, because the water, having succeeded in getting easily through the grating formed by the seat, will meet with very little resistance in moving out from under the valve. Valves free to lift to an unchecked height will often get so far away from their seats that slamming will take place at the reversal of the plunger. A pumping engine will work best when provided with sufficient valve-seat area to keep the mean velocity of the water down to about 3 ft. per sec., the lift of the valves being so restricted that they will return to the seats when the plunger approaches the end of its stroke.

5 With reference to plunger travel in conjunction with pump valve area, mentioned or inferred in the paper, the vital question is, How shall we obtain any certain plunger travel per minute: by a short stroke at many revolutions per minute, or, by a long stroke at few revolutions per minute?

6 After the water is well started through the pump valves, a larger increase in speed would be permissible than is found in practice, if it were not for the reversals at the end of the strokes. The 250-ft. per min plunger travel mentioned in the paper, would be permissible with a 60-in. stroke at 25 r.p.m., or better with a 72-in. stroke at 21 r.p.m. The pump valves would work in a very satisfactory manner, the pumps would give very good hydraulic efficiency and the engines would run smoothly. But if we should attempt to obtain 250 ft. per min. with a 30-in. stroke at 50 r.p.m. there would be a great reduction in economy, smoothness of running and general efficiency.

7 The items in Par. 7 are all within the scope of mechanical efficiency, and will be reasonably well taken care of, if the valve factor is properly attended to. The most effective method for dealing with the question of valve area, is to establish a certain satisfactory area per unit of pumpage, at some definite minimum rate of revolution as a standard. Then, for every revolution per minute above the standard rate, add a certain per cent to the standard valve area. This will give an engine of more revolutions, a greater proportionate valve area than a slower machine, thus in fast engines keeping the valves nearer to their seats than in slow ones.

8 In Par. 26, the author makes a statement, with which one feels compelled to ask issue: "———the total valve area in this type of engine need not be more than the plunger area." As already pointed

out, there is no necessary relation between the valve and the plunger area at all. The relation is only incidental, or whatever it happens to be after the proper proportions are established. A certain area of plunger, with a certain stroke, at a given number of revolutions per minute sets up a certain velocity in the water through the valve seats. A plunger of half the area, with the same stroke and at twice the revolutions per minute, will set up the same velocity of displacement, and consequently the same mathematical velocity will be required through the valve seats; although the increased frequency of opening and closing will introduce another element for consideration, which will call for a greater proportionate valve area, for the greater number of revolutions per minute. In other words, a larger plunger running slowly will require the same valve area as a smaller plunger running faster, so far as the calculated displacement and velocity are concerned. The valve area in both cases depends upon the quantity of water and the selected velocity through the valve seats, regardless of the size and speed of the plungers.

9 The spring diagram and expressions are very nicely worked out, but the differentiation is too fine for real work, and could be mostly avoided by keeping the valves closer to their seats and avoiding refinement in springs. The idea is to get away from the laboratory engine, determine the conditions to be found in a pumping station, and then meet those conditions as they really exist, rather than try to adjust the working conditions to some real although impracticable refinement in some particular factor.

10 In many pumping engines now at work, some of the details worked out very nicely on the drawing board but failed to meet the actual requirements. There are waterworks engines of the cage pump construction, in which the ends of the valve stems, with valves exactly like those shown in the paper, have been sawed off, the valves being kept in place by means of wooden wedges, just because someone who never saw the inside pump after it left the shop, did not understand the requirements involved in the care and maintenance of the machine. In one or two such cases, the cages were difficult to remove, and there was not room enough to remove the valves, with the cages in the pump chamber, by the regular method of taking off the spring guard.

Irving H. Reynolds. Mr. Nagle calls attention to two very common errors which purchasers of pumping machinery fall into when preparing specifications:

a The absurdity of specifying the ratio between plunger and valve area without other limiting clauses.

b Specifying an unnecessarily large amount of valve area. Mr. Nagle suggests as a remedy for the first, specifying *velocity* through the valves rather than a *percentage* of plunger area, and for the second, the use of lighter springs, thus enabling the valves to rise to their full lift and thereby reduce the number of valves required.

2 In regard to the first, there is an increasing tendency among engineers to specify a maximum velocity of flow through the valves rather than their area relative to the plunger.

3 Quietness of operation rather than cost is the first consideration in the design of pump valves, and the present excessive valve areas have grown from this idea. Time is also an important element in determining pump valve action; therefore, the number of reversals or valve seatings, rather than the piston speed, is the important factor, and consequently valves of small diameter and therefore of relatively low lift, have displaced the large diameters in common use a few years ago.

4 To further decrease the lift of the valves and, therefore, permit them to close quickly and quietly at high speeds, valve areas have been increased to a point where in actual operation the valves lift only a fraction of the theoretical height to which they should lift to give a full opening; in other words, large valve area is provided for the purpose of not using it.

5 If on a high-speed (high-revolution) pump the valves were fitted with light springs, permitting them to lift to their full height as suggested by Mr. Nagle, it is probable that the pump would be exceedingly noisy, as the valves would be so far from their seats at the time of plunger reversal that they would not seat until the flow through them had reversed, and this slowness in seating would be still further aggravated by the light springs employed. There is no doubt, however, that in many cases the springs used are unnecessarily stiff and on slow-speed engines the lighter springs would be found satisfactory.

6 In earlier practice, particularly with direct-acting pumps, the valve area was small in proportion to the plunger and the valves were obliged to lift nearly to their full height. In this type of pump, as the plunger speed was relatively high to nearly the end of the stroke, the valves became noisy if the pumps were operated at high speed.

7 With the general introduction of the crank and flywheel pump came higher rotative speeds and the necessity for larger valve area and smaller valve opening, i. e. lower lift, until present practice has crystallized at velocities of 3 ft. to $3\frac{1}{2}$ ft. per second through the valves, and valves of between $3\frac{1}{2}$ and 4 in. in diameter for ordinary waterworks service. In general, the best results would be obtained if engineers in drawing specifications would limit the mean velocity of water through the valves at about 3 ft. per second and the diameter of the valves to not over 4 in.

F. W. Salmon. I prefer to make these valves somewhat different from the one illustrated in the paper. I do not believe it is best to use the radical ribs of the valve seat to screw it in, but that it is better to cast small projections on the outside, as at *A* Fig. 1. This part is of such a size that an ordinary black pipe will fit neatly when properly milled out at the end, thus making a good socket wrench at a minimum cost.

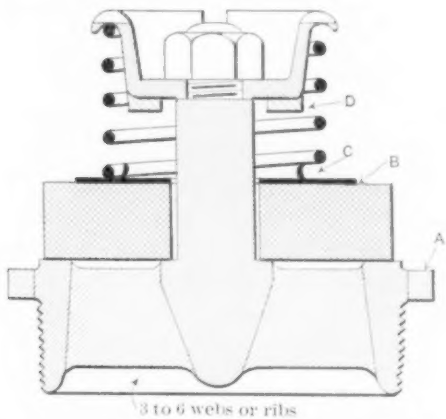


Fig. 1 Cross Section of Pump Valve, Showing Improvements Suggested by Mr. Salmon

2 I prefer to put a brass plate on the top of the rubber valve, as shown at *B* Fig. 1, and to partially punch out and turn up little projections from this plate as at *C* Fig. 1 and Fig. 2. The plate prevents the spring wearing into the top surface of the valve, and the projections keep the spring properly centered.

3 Small projections should be cast on the under side of the spring guard as shown at *D* Fig. 1 and Fig. 3, the latter being the under side

of the spring guard. If the valve is ever drawn so high as to come into contact with these projections it will still descend freely, not being in the least hindered by the soft surface of the valve forming a close contact with the smooth under surface of the spring guard, as it

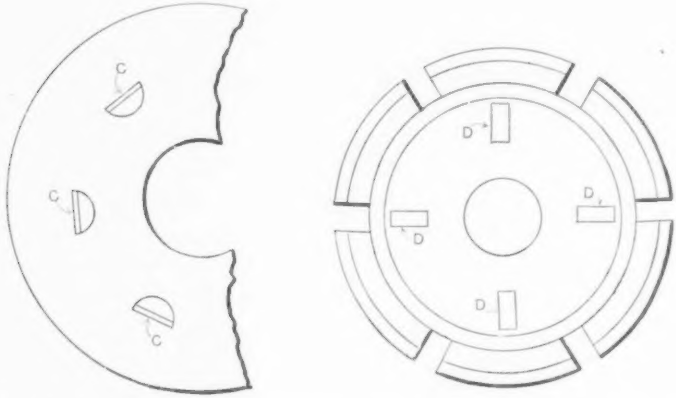


Fig 2 and 3 Showing Projections on Brass Valve Plate and on Spring Guard

is sometimes made. I consider that this is useful in cases of fast running pumps, as in such machines it is particularly desirable to have the valves seat while the crank is passing the dead center, and so a quick closing action is required.

William Kent. I hope Mr. Nagle will supplement the paper by telling us what proportion of valves and valve springs he would use for certain conditions. The paper is now largely one of criticism, and I would like to have the author make it a constructive paper. Par. 25 reads "The place to begin the study of proportions of a pump is at the spring of the valve. Make a sample spring of such diameter and length and strength as you may think desirable, and by experiment construct a diagram of its rate of compression, as in Fig. 1." This is good advice for pump designers, but other mechanical engineers are called in to confer about these points, and if Mr. Nagle would tabulate the proportions of springs suitable for pumps, and give the lifts at certain velocities of water, his paper would be more useful to these engineers.

2 The author criticises the practice of specifying the percentages of area of the valve and the pump. I see nothing very wrong

in that, provided the plunger area and the speed are also specified, as is usually done, otherwise some of the bidders will put in a small pump. In order to compel them to supply a pump large enough, we limit the velocity of the plunger; and having limited the velocity of the plunger and specified its size, we may as well say that the valve must be so many per cent of the plunger area, as to state what the velocity of the water must be. The specification is good enough, provided these additional items of plunger area and speed are also specified.

Prof. R. C. Carpenter. It is quite evident to any one familiar with hydraulics that the difficulties from the narrowing of the valve are largely inherent in the spring. If a spring could be obtained which would open uniformly with increase of pressure the troubles due to certain inertia effects which are mentioned, would disappear. This, however, merely points out the source of trouble and leaves the question open as to what shall be done. In substance, defects are merely pointed out without remedies. I would suggest, if Mr. Nagle can, that he give some of these remedies for the troubles which he has described.

E. H. Foster. Attention should be called to the fact that this paper refers to the valves of one type of pump. Many pumps of other types are built, particularly those without fly wheels, to which it is not absolutely necessary that these rules should apply. It is well known that a considerable pause at the end of the stroke of the duplex pump facilitates the closing of the valves, so that these empirical rules for lift and area must be quite different for that type of pump.

The Author. Some new matter which has come to the attention of the writer of this paper, is appended herewith. A careful study of this will answer most of the points raised in the discussion, especially the point made by Mr. Reynolds and Mr. Hague, to the effect that the maximum velocity through the valve should be limited to 3 or 4 ft. per sec. It can be assured that the formula and Table 1, quoted from Professor Bach's experiments governing the relation of spring pressure and velocity of flow to the coefficient of contraction, is correct.

TABLE I PROFESSOR BACH'S EXPERIMENTS WITH A FLAT VALVE AND A FLAT SEAT (SEE FIG. 1)

Inside diameter of valve seat d = 1.968 in. Outside diameter of valve d_1 = 2.362. Ratio of inside and outside areas, 1 to 1.44. Inside area, 3.04 sq. in.

		H = 1.27 – 1.29 ft.				H = 3.08 – 3.11 ft.	
1	M = lift of Valve, in..........	0.23	0.55	1.01	0.122	0.40	0.65
2	G = Weight of valve, lb......	2.028	2.218	2.304	4.610	5.073	5.238
3	Q = Volume of water, lb. per sec..............................		6.548	8.554	3.086	6.768	11.20
4	H = Head of water, ft........	1.29	1.29	1.27	3.11	3.10	3.08
5	W = Velocity through seat. ft. per sec........................		4.97	6.46	2.33	6.17	8.46

Calculations by Nagle

6	$\frac{m}{d}$ = Ratio lift, to diameter.	0.12	0.28	0.51	0.06	0.20	0.33
7	G = p Weight per sq. in., lb. per sq. in...........	0.666	0.728	0.760	1.516	1.668	1.723
8	V_g = Velocity due to p, ft. per sec...............	9.55	9.47	9 07	14.70	14.72	14.35
9	V_h = Velocity due to H, ft. per sec................	9.12	9.12	3.92	14.14	14.10	14.07
10	Ratio of $\frac{V_g}{V_h}$..................	1.04	1.04	1.02	1.04	1.04	1.02

Line 7 is obtained by dividing the weight G, given in line 2, by the area of d, or 3.04 sq. in.

Line 8 is obtained by the aid of Table 2, where opposite the value of $\frac{m}{d}$ is found the coefficient and formula. For example: taking the first case of a lift of 0.23 in., or a percentage of 0.12 of the diameter 1.968 in., we find by interpolation in Table 2, the formula, $V = 1.17\sqrt{100\,p}$, or $V = 1.17 \times 8.16 = 9.55$ ft. per sec.

Line 9 is obtained from the fundamental hydraulic formula $V = 8.025\sqrt{H}$, when H is the head in feet and V the velocity in feet per second. For example, in the first case cited we have, H = 1.27 ft. $V = 8.025\sqrt{1.27}$, or = 9.12 ft. per sec.

Line 10 is self-explanatory, and is introduced as a check upon the work and formulæ, as if correct, it should be unity. The slight deviations are due to the various decimals not being carried far enough, but they are carried far enough for all practical purposes.

TABLE 2

1	$\frac{m}{d}$ =	....	0.05	0.10	0.15	0.20	0.25	0.30	0.35	0.40	0.5d
2	u =	....	0.65	0.60	0.56	0.53	0.50	0.47	0.44	0.41	0.37
5	p =	0.67	0.69	0.72	0.74	0.77	0.80	0.83	0.86	0.89	0.92 $\frac{V^2}{100}$
6	v =	1.22	1.20	1.18	1.16	1.14	1.12	1.10	1.08	1.06	$1.04\sqrt{100p}$

Line 1 $\frac{m}{d}$ is the actual rise, or lift, of the valve, divided by its inside seat diameter.

Line 2 u is the coefficient of contraction at the point of discharge with a given lift.

Line 4 V is is the velocity of the issuing stream at the point of discharge in feet per second.

Line 5 p is the pressure in pounds per square inch and is found by dividing the weight of the valve (in water), plus its spring pressure in pounds by its inside seat area in square inches.

Line 6 v is the velocity of the issuing stream per second.

2 Let us apply the formula to the 3 ft. per sec. assumption. For a lift of, say, $0.20 \times$ diameter,

$$p = 0.77 \times \frac{V^2}{100} \text{ or } = 0.07 \text{ lb. per sq. in.}$$

of inside seat area. Such small spring pressure is out of all proportion to what common practice has established, which is from 0.30 to 0.60 lb. at the initial and 0.75 to 1.50 lb. at the full lift. The formula for the resulting velocities is very simple. Suppose we solve for four spring pressures of, say, 1.50, 1.25, 1.00 and 0.85 lb. at full lift, and 40 per cent, or 0.60, 0.50, 0.40 and 0.34 lb. at the initial point. at a lift of $0.20 \times$ diameter, the formula would be

$$V = 1.14 \sqrt{100 \times p}$$

and the velocities for

$$\begin{aligned} p = 1.50 \text{ lb.} \quad & v = 13.96 \text{ ft. per sec.} \\ 1.25 \text{ lb.} \quad & v = 12.74 \text{ ft. per sec.} \\ 1.00 \text{ lb.} \quad & v = 11.40 \text{ ft. per sec.} \\ 0.85 \text{ lb.} \quad & v = 10.51 \text{ ft. per sec.} \end{aligned}$$

The coefficient of contraction would be 53 per cent in each case.

3 It is plain, therefore, that we are far from realizing four feet per second with our present spring practice.

4 To Mr. Reynolds: The writer did not mean to lighten the springs abnormally, in fact, 0.45 lb. to 0.50 lb. initial is probably light enough, but if they could be made somewhat longer, so as not to tighten up too rapidly, it would seem to be desirable.

5 To Mr. Kent: The formulæ given by Professor Bach are a very great addition to our knowledge of pump-valve action. Within the limits prescribed, we have now a safe guide for valve construction. What it should be for other numbers of revolutions and plunger velocities, I am not able to formulate. Professor Haeder goes into that phase of the problem, but as his theory is not confirmed by extensive experience, I do not take it up in this paper.

ADDENDUM TO PAPER

6 In Par. 15 of the paper is given a formula for ascertaining the lift of a pump valve, from which was omitted, as stated, the coefficient of contraction. Not knowing the value of this coefficient with certainty, the writer hoped the information would be supplied in the dis-

cussion. The omission was not referred to, however, and he is now able to supply it himself.

7 In a German book on pumps and pump valves by Herm. Haeder, Duisburg, the subject is treated in an exhaustive manner. The actual coefficients of contraction are given, with the results of impact upon the valve, based upon experiments by Professor Bach. In what follows reference is made only to that part which bears on the subjects of flat valves and flat seats, of which the inside and outside areas bear the ratio of 1.00 to 1.44. The notations were originally in French, but in what follows have been transformed into English units.[1]

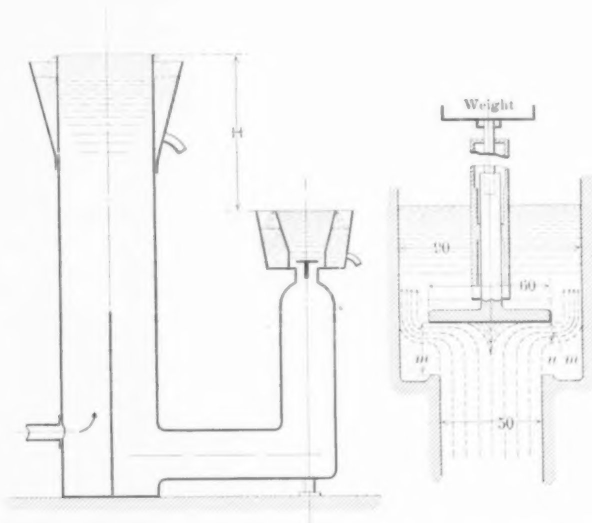


FIG. 1 APPARATUS USED BY PROFESSOR BACH FOR DETERMINING COEFFCIENT OF CONTRACTION

8 Fig. 1 shows the apparatus used by Professor Bach. Table 2 (Haeder 261) gives the original data and results obtained and some calculations of my own, the better to elucidate the subject.

9 Fig. 2 and Fig. 3 (Haeder 110 and 110a) show a valve closed and one open, with the respective formulæ for the two positions of the valve, giving the values for velocity or pressure in the two extreme positions. "Open" signifies a lift of one-half the diameter, which, needless to say, is far beyond American waterworks practice.

[1] The original tables in French units can be referred to in the author's manuscript on file in the rooms of the Society.—EDITOR.

10 Table 2 (Haeder 213) gives the values of v and p for the intermediate positions of the valve, and also the value of "u", the all-important coefficient of contraction at all positions. Use this table to ascertain the velocity v of the water through the valve opening and also the coefficient of contraction u at the same point.

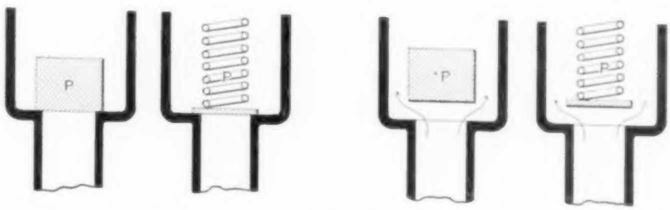


FIG. 2 AND FIG. 3 SHOWING, RESPECTIVELY, A VALVE OPEN AND A VALVE CLOSED. THE FORMULAE FOR THESE TWO POSITIONS ARE AS FOLLOWS:

V = VELOCITY IN FEET PER SECOND P = POUNDS PER SQUARE INCH

VALVE CLOSED	VALVE OPEN
$V = 1.22 \sqrt{100\,P}$	$V = 1.04 \sqrt{100\,P}$
$P = 0.67 \dfrac{V^2}{100}$	$P = 0.92 \dfrac{V^2}{100}$

11 We can now say that we have a practically correct formula for ascertaining the volume of water discharged through a flat disc pump valve of a certain diameter, an assumed lift, and a certain tension of spring.

12 Throughout all the following calculations a maximum lift of valve of 0.15 d, is taken, leaving the reader to make for himself other assumptions of lift and the consequent calculations. Various tensions of springs will be taken, to illustrate the importance of giving more attention than heretofore to the strength and length of springs.

13 Take, for examples, the same dimensions of pump and valve as those used in Par. 16. Formula 3 (Par. 15) would now be better expressed in terms of N, the number of valves, than assuming the number of valves and solving for the lift L. The formula would then read,

$$N = \frac{P \times V_m}{C \times L \times u \times V_m}.$$

Applying this formula to the three different strengths of springs before used, we get the following results:

14 First, ascertaining the velocity through the valve by the aid of Table 2, the spring tensions were as follows;

Case 1: initial, 0.60 lb.; final 1.55 lb. per sq. in.
Case 2: initial, 0.40 lb.; final 1.03 lb. per sq. in.
Case 3: initial, 0.30 lb.; final 0.77 lb. per sq. in.

The formula for the velocities due to these final pressures at a lift of 0.15 d, are

$$\text{Case 1: } V = 1.16\sqrt{100 \times 1.55}, \text{ or } V = 14.41 \text{ ft. per sec.}$$

$$\text{Case 2: } V = 1.16\sqrt{100 \times 1.03}, \text{ or } V = 11.77 \text{ ft. per sec.}$$

$$\text{Case 3: } V = 1.16\sqrt{100 \times 0.77}, \text{ or } V = 10.18 \text{ ft. per sec.}$$

The coefficient of contraction in each case is $u = 0.56$. Substituting these values in Formula 4, we have,

$$\text{Case 1: } N = \frac{908 \times 6.67 = 6056}{10.53 \times 5.625 \times 0.56 \times 14.44} = 127$$

$$\text{Case 2: } N = \frac{6056}{3.317 \times 11.77} = 155$$

$$\text{Case 3: } N = \frac{6056}{3.317 \times 10.18} = 180$$

15 Let us make a similar calculation for springs of the same initial strength, but longer, so that they will tighten only one-half as much in their nine-sixteenths lift. Then the first spring final tension becomes 1.08 lb., the second spring 0.72 lb., and the third spring 0.53 lb.; and the velocities become

$$\text{Case 1: } V = 1.16\sqrt{100 \times 1.08} = 12.05 \text{ ft. per sec.}$$

$$\text{Case 2: } V = 1.16\sqrt{100 \times 0.72} = 9.84 \text{ ft. per sec.}$$

$$\text{Case 3: } V = 1.16\sqrt{100 \times 0.56} = 8.68 \text{ ft. per sec.}$$

and solving for N in formula 4, we have

$$\text{Case 1: } N = \frac{6056}{3.317 \times 12.05} = 152$$

$$\text{Case 2: } N = \frac{6056}{3.317 \times 9.84} = 186$$

$$\text{Case 3: } N = \frac{6056}{3.317 \times 8.68} = 210$$

16 To calculate the loss of efficiency for these different springs let us take the mean pressure on the springs to be the initial, plus two-thirds of the increase, and twice this for the two strokes, and this sum must be divided by the total pump head, say 80 lb., to obtain the loss of efficiency. We would then have

Case 1: $[0.60 + (1.55 - 0.60)\ \tfrac{2}{3}] \times 2 \div 80 = 3.06$ per cent
Case 2: $[0.40 + (1.03 - 0.40)\ \tfrac{2}{3}] \times 2 \div 80 = 2.05$ per cent
Case 3: $[0.30 + (0.77 - 0.30)\ \tfrac{2}{3}] \times 2 \div 80 = 1.50$ per cent

With stronger springs, we would have

Case 4: $[0.60 + (1.08 - 0.60)\ \tfrac{2}{3}] \times 2 \div 80 = 2.30$ per cent
Case 5: $[0.40 + (0.72 - 0.40)\ \tfrac{2}{3}] \times 2 \div 80 = 1.50$ per cent
Case 6: $[0.30 + (0.53 - 0.30)\ \tfrac{2}{3}] \times 2 \div 80 = 1.15$ per cent

Grouping these figures for better comparison, we have Table 3.

17 We have now, in Table 3, figures which enable us to study pump valve constructions in an intelligent manner. The formulæ given enable us to construct a similar table for any other assumed dimension of plunger and its velocity, height of lift of valve, or spring tension.

18 In conclusion the writer wishes to say that now, for the first time in the history of the modern high-duty pumping engine, we have a formula for designing a pump valve that is scientifically correct, and one based upon hydraulic experiments carefully made by a competent authority. The subject seems important enough to bear repetition in grouping the previous instructions, as follows:

19 Find the area of the plunger in square inches, and the maximum speed of the plunger in feet per second. The latter is found by multiplying the stroke in feet by the maximum number of revolutions per minute, multiplying this result by 1.60, to reduce it to its maximum velocity (the crank velocity), and dividing by 60 to reduce it to feet per second. This product, algebraically expressed by $P \times V_m$ in Formula 4, becomes the numerator of the equation.

20 Determine the size of the pump valve-seat and its net area between the ribs, whether the valve bears on the ribs or not; that will be the inside area of the valve against which the impinging stream acts.

21 Decide what lift of valve you intend to have. American water works practice is from 0.10 to 0.20, the diameter of the inside of the outer seat. This lift is designated by L in the formula.

22 Decide what spring pressure you will have, both at the beginning and at the full lift. This spring pressure is expressed in pounds

per square inch of inside valve area and usually runs from 0.30 to 0.60 lb. per square inch at the beginning of the lift, and it ought not to be quite double this amount when the valve is full open to its stop. It will be this final tension, plus the weight of the valve in water, designated by p in Table 2, that will be the determining factor for the velocity of the issuing stream. To illustrate, if the final pressure be 0.81 lb. per sq. in., with a lift of 0.15 d, the equation (see Table 2) for $V_m = 1.16\sqrt{81} = 10.44$ ft. per sec.

23 The discharging area is the net circumference of the inside valve diameter C, taking out the ribs whether they support the valve or not, multiplying this by the actual lift and this product by the coefficient of contraction u, as found also in Table 2, which, for the lift cited, is 0.56.

24 Algebraically expressed, these factors become the denominator in the Formula 4,

$$N = \frac{P \times V_{\mathrm{m}}}{C \times L \times u \times V_{\mathrm{m}}}$$

TABLE 3 $\frac{9}{16}$ IN. LIFT

PLUNGER 34 IN. DIAMETER BY 5-FT. STROKE BY 25 R.P.M. MAXIMUM VELOCITY = 6.67 FT. PER. SEC. VALVES $3\frac{3}{4}$ IN. INSIDE DIAMETER. NUMBER OF VALVES, SPRING TENSIONS AND PUMP EFFICIENCIES.

	Initial and Final Spring Tension Pounds	Valve Seat Area Per Ct.	Number of valves	Lift of valves Inches	Maximum Velocities Feet per Second			Loss of Efficiency Per Ct.
					Plunger	Valve Seat	Valve	
1	2	3	4	5	6	7	8	9
1..........	0.60 to 1.55	112	127	$\frac{9}{16}$	6.67	5.96	9.44 – 14.44	3.08
2..........	0.40 to 1.03	137	155	$\frac{9}{16}$	6.67	4.87	7.71 – 11.77	2.05
3..........	0.30 to 0.77	159	180	$\frac{9}{16}$	6.67	4.20	6.68 – 10.18	1.50
			Longer Springs					
4..........	0.60 to 1.08	134	152	$\frac{9}{16}$	6.67	4.98	9.44 – 12.05	2.30
5..........	0.40 to 0.72	164	186	$\frac{9}{16}$	6.67	4.07	7.71 – 9.84	1.52
6..........	0.30 to 0.53	185	210	$\frac{9}{16}$	6.67	3.60	6.68 – 8.68	1.15

Column 3 is obtained by multiplying the number of valves by the net area through the seat (8.00 sq. in.), and finding its ratio to the plunger area.

Column 5 is taken at $\frac{9}{16}$ in. = $(0.15 \times d)$ in all cases.

All the other data have been already explained.

AN EXPERIENCE WITH LEAKY VERTICAL FIRE-TUBE BOILERS

By F. W. Dean, Published in The Journal for October

ABSTRACT OF PAPER

This paper discusses the difficulties experienced with some large vertical boilers, somewhat over 10 ft. in diameter, and containing over 6000 sq. ft. of heating surface. The boilers leaked very badly very soon after being started and nothing that was done improved their condition until the water legs were lengthened from 2 ft. to 7 ft. $2\frac{3}{4}$ in. The boilers were raised 5 ft. $2\frac{1}{4}$ in. Before they were raised the lower ends of the tubes would cover with very hard clinker and become stopped up. This clinker could be removed only by cutting it off when the boilers were cold. After the boilers were raised, a light clinker that could be blown off formed about the tubes; by removing this by blowing every three or four hours the leaks were stopped and they have never returned.

The trouble varied with different kinds of coal Each boiler had been run constantly at over 1000-h.p. and the economy seemed to be about the same no matter what the power was. So far it has been difficult to obtain good combustion, but the heat-absorbing power of the boilers is admirable. The experience with these boilers indicates that there is no ordinary limit to the size of a vertical fire-tube boiler.

DISCUSSION

Reginald P. Bolton. It appears to me that this design of boiler was an invitation to the trouble that followed, and it is only necessary to go back into the experience of other people to find out that others have suffered in the same manner. If the view of the boiler as presented in the paper is turned horizontally, and it is imagined that it is a locomotive boiler cut off short, it will be seen that there is no combustion chamber whatever in it. This boiler was to be put to a service which might call for a rate of combustion in the furnace which would demand double its rated capacity output, so that the double aggravation of a very small combustion chamber and very large rate of combustion, was present.

2 The design of the boiler is radically defective in two important points, namely, the tubes are entirely too long, and the combustion space was entirely too small. It is now very nearly half a century ago that the experiments of Geoffroy and Petiet demonstrated the futility of unduly lengthening fire tubes. These experi-

ments demonstrated the rapid reduction in efficiency due to length of tubes, under various conditions of draft and rates of fuel consumption. Almost precisely the same conditions were tested as in the author's boiler, as follows:

3 A consumption of fuel exceeding 50 lb. per square foot of grate, under an air pressure of 2.36 in. with the following results:

	Evaporation per Sq. Ft. Lb.
Fire-box plate	23.5
First three feet of tubes	5.4
Second " " " "	2.5
Third " " " "	1.33
Fourth " " " "	0.83
Fifth, three feet evaporated only	0.48
Sixth " " " "	0.3

The last two were found by extending the curve.

4 An examination of these results might have dissuaded the author from the mistake of designing the boiler with such a length of tube, involving not only inefficiency, but the evident concomitant of leakage as a result of expansion and contraction. Apart from the other defective feature, the boiler could have been shortened so as to reduce the tubes at least five feet in length, and would no doubt have given better efficiency as a result.

5 The general type of the boiler possesses nothing new or original unless we may so regard the restricted combustion chamber, by which the tube plate was brought within seven feet of the grate, allowing a total capacity of only 535 cubic feet for the fire and for the gases of combustion.

6 A very simple computation of the results of the combustion of 40 lb. of coal per square foot of grate area, will show that the volume of products of combustion would be so great, that only an excessively heavy draft could force them through the combustion chamber and tubes, and that incomplete combustion was bound to result.

7 The addition of $5\frac{1}{2}$ ft. to the height of the chamber, which was arrived at only after three years experience with this boiler, nearly doubled the effective space for combustion, and also removed the ends of the tubes from the direct action of the blast. It may be observed that a Dutch oven would have afforded equal results, at perhaps less expense.

8 The reason for the adhesion of molten clinker to the ends of the tubes, need have presented little difficulty, in the light of past

experience, since the ends of the tubes were placed so close to the fire. This result developed in the fire-tube boilers of H. M. S. Polyphemus nearly thirty years ago, and when found in the boilers of locomotives is due to precisely the same cause.

9 It will be noticed that the best of the tests which were made after the change of combustion chamber was effected, is that in which the rate of fuel consumption is least.

10 I agree with the quoted conclusion of the second boiler expert, referred to in Par. 6, and am at a loss to understand why such an opinion, thus expressed, was regarded as unsatisfactory. It may be hoped that the paper may stand as a warning signal to other designers. It requires a great deal of courage to present a paper of this kind, and the author should be thanked for bringing forward a record of a failure so that we may profit by the facts.

William Kent. I join with Mr. Bolton in praising Mr. Dean's courage in bringing forward a report of his failure, and I regret that some eight or ten years ago I did not bring forward a record of another similar failure, not my own, but that of some other man, which might have prevented Mr. Dean's bringing forward a record of this failure. The New York Steam Company bought a boiler for their Greenwich Street Station to go in a very small ground space. It was a very large plain vertical cylinder boiler, eight or ten feet in diameter, full of tubes about 20 ft. long, and was rated at 1000 h.p. It had not been in use more than a week or two when it began to leak. There was no way to clean the flat tube sheet or to clean the tubes of scale, and the boiler was condemned and taken out.

J. C. Parker. The reason that the tubes leaked was that when the boiler was set close to the grate the tube ends were subjected to wide fluctuations in temperature. The flow of air through a chain grate increases toward the rear end, and where the boiler was set higher there was more mixing of the hot and cold currents and, consequently, less fluctuation in temperature.

2 The clinkering of the tubes would naturally increase the trouble because of the concentration and increased friction of the gases in the tubes that remained clear.

Orosco C. Woolson. This discussion has brought out the important fact that perfect combustion should take place before the gases reach the tubes or shell of the boiler.

2 I have been somewhat surprised in my travels among the cotton and woolen mills of the Eastern States where the management have large experience in cotton spinning but are limited in personal experience regarding what constitutes the production of the highest calorific value of a pound of bituminous coal. One man of large experience in mill work wanted his furnace fire directly under the tubes of his vertical boilers, and gave me his reasons. I told him that I would guarantee him better results if he would discard the idea that the area immediately under and against the tube sheet should act as a combustion chamber. Let combustion take place entirely before it reaches the tube sheet and the results will be much more satisfactory.

3 Secondly, as to the tubes filling with vitrified slag or any other residuum of combustion, I would suggest that such deposit should be made to take place under a fire arch, where it will adhere to the crown and serve a useful purpose by forming a refractory coating. This practice is becoming popular, and more so today than ever before. It is my opinion that by completing combustion under a properly constructed arch within a properly constructed combustion chamber, the products of this combustion will be sent to the boiler in the form of what we will term "caloric ether" and not a mixture of its original constituents which play no useful part, under the circumstances, in producing or maintaining heat, but rather are subject to ready condensation.

A. A. Cary. In my experience with vertical fire-tube boilers I once found a boiler containing shorter tubes and of a greater diameter than are ordinarily found in the Manning type. The fuel used was a moist anthracite coal, and there was a natural draft of more than one inch of water in the smoke box over the boiler. The draft could not be regulated, due to the previous burning out of the steel plate butterfly damper. The partially burned furnace gases passed rapidly through the vertical tubes and ignited above the top tube sheet, thus causing the destruction of dampers and the steel breeching, to say nothing of the reduced evaporation in the boiler due to this waste of heat.

2 The trouble was remedied by placing the grates at a greater distance from the lower tube sheet and arranging baffles in the combustion chamber so as to insure the more complete combustion of the gases before they entered the tubes. A cast-iron plate damper replaced the former one of steel plate, and no further trouble has since been experienced.

3 In another case, the question came up as to the advisability of applying a special automatic furnace, using bituminous coal and producing very high temperatures, under boilers of the Manning type. An arrangement which has been used in New York City for a number of years was suggested and successfully applied.

4 Fire-bricks, piled on edge with open spaces between the bricks, were arranged a short distance beneath the lower tube sheet. This checker work of bricks filled the entire space beneath the boiler, the openings between the bricks at the center being very much reduced, so as to cause a decreased flow of gases directly under the center of the overhead tube sheet. By this means, a very even distribution of temperature was secured over the entire area of the lower tube sheet with a slight reduction of heat delivery at its center, the most sensitive portion of the whole tube area.

5 The author mentions inefficient combustion, which is indicated by the comparatively low percentage of CO_2 and high percentage of O, shown in Table 1. As the higher temperatures are secured by the most complete combustion with the least excess of air, the question arises, why should such destructive results follow such inefficient furnace conditions?

6 Pyrometric testing with gas analyses have taught me that when a furnace is being operated inefficiently, very high temperature may be found in one part of the furnace while at the same time a comparatively low temperature may exist in another part. This may lead to the simultaneous impingement of gases of very different temperatures upon various parts of the lower tube sheet, setting up destructive strains and contributing to such troubles as have been described by Mr. Dean.

7 The lower tube sheets of boilers of the Manning type are very sensitive, especially towards the center of the sheet where the water seems to penetrate with great difficulty, thereby failing to keep this portion of the heating surface constantly wet.

8 Concentration of heat due to concentration of combustion and lack of space for this small volume of high-temperature gas to diffuse itself throughout the entire mass of furnace gases before they reach the tube sheet, is bound to cause trouble, especially when this highest temperature is concentrated against the center of the tube sheet on the inner surface of which there is apt to be little or no water. After the center of this sheet loses the supporting effect of the center tubes, acting as stays, the surrounding tubes are very apt to follow.

9 Concerning the low efficiency of the furnace referred to in

Par. 13, there should be no trouble in remedying this fault. A properly conducted furnace test (apart from the boiler) with pyrometers, gas analyzing apparatus, etc., will show just where the trouble exists and will point out the needed changes as well as the limitations under which this type of stoker can be worked with the different grades of fuel used.

Prof. L. P. Breckenridge. One of the speakers said that the highest temperature in a boiler furnace is directly over the fire. This is not always so. We have measured the temperature twenty feet from the fire and found it higher. It depends on the volatile content of the fuel and whether the flame has been supplied with a sufficient amount of air early in the process of combustion. It is this that determines whether the high temperature point is ten feet or twenty feet away. Many times in our experiments in the St. Louis boiler trials we have seen that every time the furnace door was opened the temperature at the rear end of the combustion chamber went up, because when more air was admitted the combustion was better and the temperature increased.

2 For experiments concerning the transmission of heat through a boiler tube, it occurs to me that Mr. Dean has designed one of the most satisfactory laboratory boilers I have seen. There has been much discussion of late on the heat transferred through a boiler tube, as influenced by the velocity of the gases passing through the tube. This boiler with its large number of tubes would be just the type with which to make a test on this point. I wish Mr. Dean would burn a large amount of coal per square foot of grate in this boiler furnace, using, first, all the tubes, and secondly, only one-half the tubes. If the same amount of coal was burned in each case the velocity of the gases through the tubes would be twice as great in the second case, and it would be interesting to know the relative amounts of heat transferred.

3 I hope that some time we may take up the question of the burning of fuel, making a distinction between the economical performance of the boiler and of the furnace. We have reached a time when we can intelligently discuss these questions separately. Anthracite coal, on account of its high fixed-carbon content, is burned mostly on the grate itself. When burning semi-bituminous coal, with 18 to 20 per cent volatile content, a large combustion chamber is required, and as the volatile content increases the size of the combustion chamber must be increased. When burning bituminous coal, with

40 per cent volatile content and 20 per cent ash, the fuel actually burned on the grate is small. The grate supports the fuel and some coal is burned there, but it is in the combustion chamber that we burn fully one-half the combustible part of our fuel. It is evident that more attention must be given the proportions of our combustion chambers when burning high-volatile coals, and especially at high rates of combustion.

Prof. A. M. Greene, Jr. In London Engineering for October 22 and November 5, 1909, appeared an article by Professor Dalby, in which he summarized a number of articles referring to heat transference through plates. I would commend the article to the attention of all the members of the Society interested in this matter.

2 In London Engineering for Feb. 1909, Professor Nicholson described experiments showing clearly that only a small part of the possible heat transmission through plates is utilized. I mention this to call attention of the members to the fact that some data are available on this subject. In this article are given the formulæ for heat transmission which may by compared with the results of German Experiments recently completed at Dresden (Zeit. des. Verein Deutscher Ing., October 23, 1909).

William Kent. In another issue of London Engineering, a correspondent showed that the idea of high speed of the gases being favorable to combustion was negatived by the Lancashire boiler, in which the flues are very large and the speed of the gases low, yet the economy is as high as in any other boiler.

Reginald P. Bolton. It is mainly a question of the difference in temperature between the inside and outside of the heating surfaces. The lower the temperature of the feed water, and the higher the temperature of the fire, the greater will be the efficiency of the boiler.

E. D. Meier. I find myself in substantial agreement on some points with all the gentlemen who have spoken. I want to say for Mr. Dean, that he is correct in his conclusion that the precipitation which occurs at the bottom of the tubes has a great deal of influence on the overheating of the tube sheet. The other causes which were mentioned are also true, but there is no doubt an accumulation of carbon there. I do not know whether Mr. Dean preserved any of the

precipitate or stalactites, but I believe a large part of it was unconsumed carbon, which will remain at a high temperature for some time.

2 I am reminded of an experience which I had with water-tube boilers at the Chicago World's Fair. I think there were ten different makes of water-tube boilers, most of them sub-horizontal, but some of the vertical-tube type and some of the bent-tube type. We were burning crude oil, and all the boilers suffered from the same causes,—every one lost tubes by burning out. Some were careful enough to shut down a boiler as soon as they noticed the blisters on the tubes.

3 The boilers which I had at Chicago were afterward placed in the midwinter fair at San Francisco, and were fired with California crude oil for seven months without a tube being lost. These boilers were afterwards sold with the condition that if the customer found any tube damaged it would be replaced, but not one was found to be burned. That bears on the subject mentioned by Mr. Dean. The trouble we found was this: The oil is supposed to be atomized in the burners, but this is not always the case. Little slugs of oil would fly up and adhere to the tube, and would spread and slowly carbonize. They would not burn, because no air could get to them. One little spot, a half inch in diameter, would become red hot in spite of all the circulation of water, and would ultimately burn out and make a blister.

4 When the boilers were installed in California, the oil burners were placed lower and were directed downward so that the jet would strike the bottom of the combustion chamber at a distance of six feet from the front, hence there was no chance of oil striking the tubes. Perfect combustion was obtained, and on one occasion one of the boilers was forced so hard that a picture was taken of the inside of the furnace by its own heat. I have that photograph still, to show what can be done. One can see a perfectly white heat and not a single blister on the tube. In Mr. Dean's case carbon was deposited and became incandescent, and gave an intense local heat on some point, which accounts for the failure of the tubes at such point.

5 In regard to the combustion chamber, I agree with Professor Breckenridge. I have always been a believer in a large combustion chamber, and one of my early experiences in that direction was when in charge of a plant having two horizontal tubular boilers, using Illinois coal. At that time everybody in the Mississippi Valley believed in river practice. The boilers, engines and dimensions of pipes, etc., were according to river practice. The boilers were set with the grate twelve inches from the bottom of the shell. I

raised them to thirty inches, and I was told I would not get any heat, but I got better results, and the boilers lasted longer. The increase in the distance from the fire to the shell was a great advantage, and, of course, incidentally I increased the efficiency of the boiler.

David Moffat Myers. In Table 4 of my paper on Tan Bark as a Boiler Fuel, during the efficiency test the temperature inside the furnace was 1100, the temperature in the combustion chamber, under the boiler, was 1475, the flue temperature was 493, and the thermal efficiency was 71.1 per cent.

2 These figures prove that under conditions of good efficiency it is quite possible to have a higher temperature at some distance from the fuel than close to it. The combustion of the gases is simply retarded to a later point of their travel. This might be caused by the combination of a high velocity of draft with a moderate air supply, so that the oxygen does not come into sufficiently intimate contact with the fuel gases in the primary combustion chamber, that is, in the furnace proper. In the case quoted, the CO_2 ran almost uniformly at about 12 per cent, the O between 6 and 7 per cent, with no determinable CO.

A. Bement. In the boiler which Mr. Dean describes, I like the scheme of having the rear end of the chain grate exposed so that it is accessible. The capacities obtained with these boilers are very large; the strength of draft, however, is somewhat too much for an ordinary chain-grate fire. It is my experience that chain grates are not proportioned so that it is possible to carry the requisite thickness of fire for a draft such as existed in this case. I think this will account for the low percentage of CO_2 in the combustible gases, and in this is found the reason why the efficiency was not higher.

2 I would attribute the leaking of the tube ends in the head over the fire to another cause than that given. Considerable experience in similar cases leads me to believe that the trouble is caused by excessive heating on the delicate tube ends in the flue sheet. There are two thicknesses of metal to be penetrated before the heat reaches the water; also the opportunity for water to enter among the tubes and to flow freely over the heated parts is rather restricted. When the ordinary return tubular boiler is set with a fire under the shell, a large portion of the heat flows through the shell, with the result that the temperature of the gases is much reduced, so that by the time they impinge upon the tube sheet, their temperature is low enough so that no damage results.

3 A case of trouble of this kind is illustrated by Fig. 1 and Fig. 2, the first showing a return tubular boiler set against an enclosed fire-brick furnace, in which the gases first impinged upon the tube sheet, passing through the tubes to the other end of the boiler, thence find-

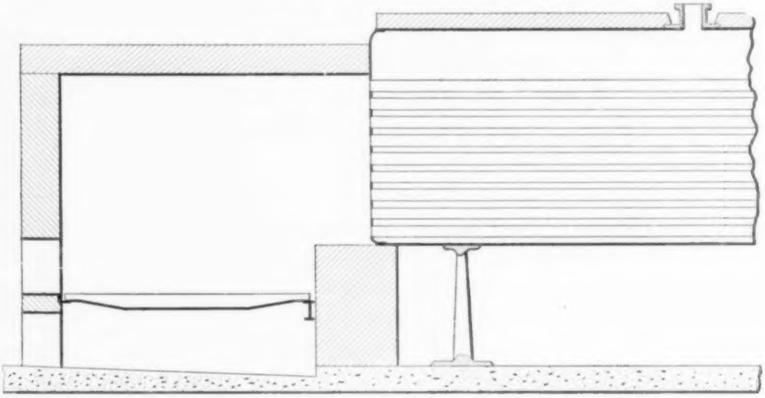


FIG. 1 SETTING OF A FIRE-TUBE BOILER IN WHICH THE TUBES LEAKE

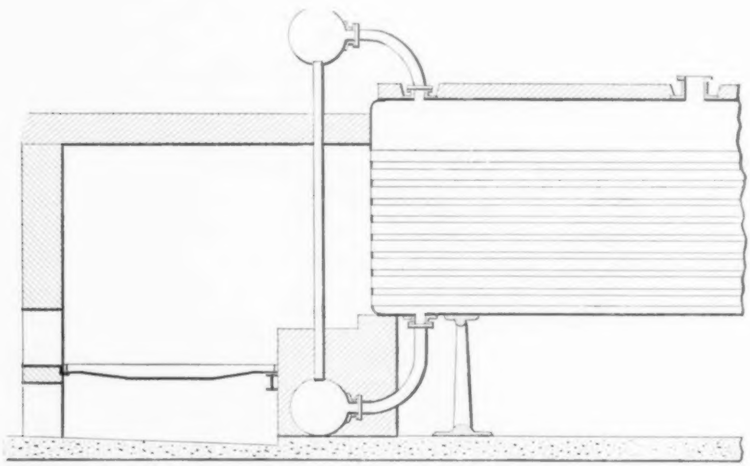


FIG. 2 SHOWING WATER LEG TO LOWER TEMPERATURE OF GASES IMPINGING ON TUBE SHEET

ing exit by way of a chimney attached thereto. When these boilers were put at work immediate and very serious trouble resulted with the tube ends; they leaked very badly, the bead getting out of shape and springing away from the sheet. By means of a little door in the

side of the furnace one could see the water squirting from every tube, and running away from the setting on the floor in a large-sized stream.

4 A remedy was effected in this case, as shown by Fig. 2, by mounting above and below the furnace a drum which extended crosswise of the setting, and connected by vertical 4-in. boiler tubes as indicated; each of these drums being in communication with the boiler, allowed circulation of water and steam. With this scheme the gases first pass between these vertical tubes, which are set closely together, with the result that there is a considerable reduction in the temperature of the gases before they came in contact with the end of the boiler tubes.

5 Another case of this character was remedied by carefully cleaning off the end of the boiler and coating it with an asbestos cement, which was rounded over and into the boiler tube openings in such a way that the flue sheet was entirely protected. This covering lasted about three months, after which it was necessary to renew it. As it was a house-heating boiler, two renewals a season served until the boiler plant was dismantled. The cure of the trouble with the boiler having the extended water leg, as shown in Fig. 2, is due in my opinion to the added heat-absorbing surface in the deeper leg, as it operates to abstract a much larger quantity of heat from the gases before they came in contact with the tube ends, than did the boiler before alteration.

The Author. Replying to Mr. Bolton's remarks, I have heard of the experiments which he quotes in regard to the rate of evaporation of different portions of the length of a tube, but I am not at all impressed with them as a guide. It is well known that the first surface that receives heat gives the greatest rate of evaporation and leaves less for the remaining su.face to do. Attention to this to the extent apparently advocated by Mr. Bolton would lead to an absurd result, for one might go on indefinitely shortening tubes. It should be remembered that only 16 ft. of the 20 ft. length of tubes are in contact with water, the remainder being for superheating.

2 Apparently Mr. Bolton believes that it is known how long tubes should be. I do not think that this is known, for the reason that a boiler must undergo a wide range of duty: a short tube would do for light work and a long one would be necessary for heavy work. Many vertical boilers with 2½ in. tubes 20 ft. long have been used successfully for years and they are still being built. Mr. Bolton would evidently prohibit increasing the size of a boiler by increasing the

length of tubes, and would recognize only an increase in diameter as a means of increasing size. To my mind this is illogical and not consistent with the teaching of successful practice.

3 Mr. Bolton speaks of the small combustion chamber as the boiler was first installed, but he ignores the hundreds, if not thousands, of vertical boilers with less combustion chamber space. I believe that I am the only person who designs vertical boilers with the crown sheet as much as 8 ft. above the grate, and this I have been doing for

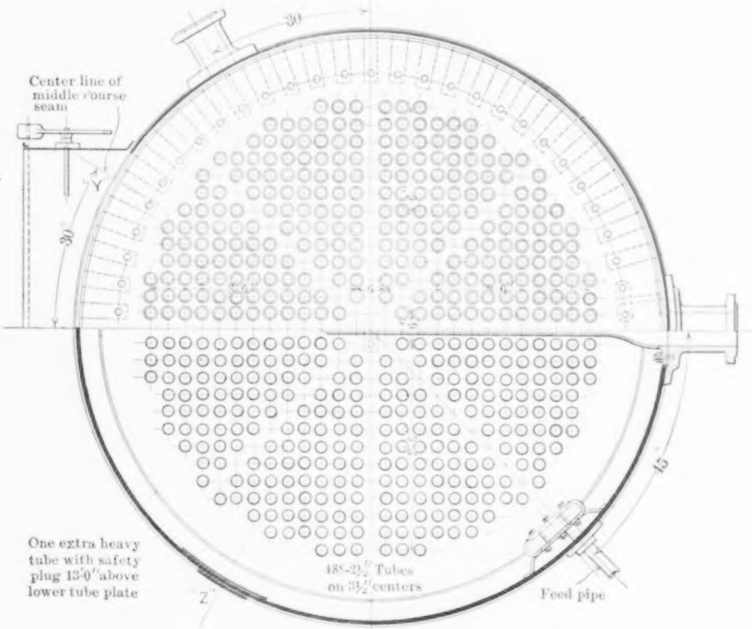


Fig. 1 Cross Section of Vertical Fire-Tube Boiler Designed by the Author

many years. In regard to the Dutch oven in front of these boilers, it would have wholly defeated the object of using vertical boilers. Besides it would have added undesirable brick work.

4 Mr. Bolton easily accounts for the lack of economy of the boiler, but ignores the perfection with which it absorbs heat. I believe the lack of economy to be wholly due to want of air, and when this is supplied and properly distributed the economy will be satisfactory. This would be equally true if the combustion chamber were much longer. The locomotive boilers tested at the St. Louis Exposition by the Pennsylvania Railroad have very little combustion chamber

space, and the excellent economy is due to the proper admission and distribution of air. In regard to the economy of the boilers under

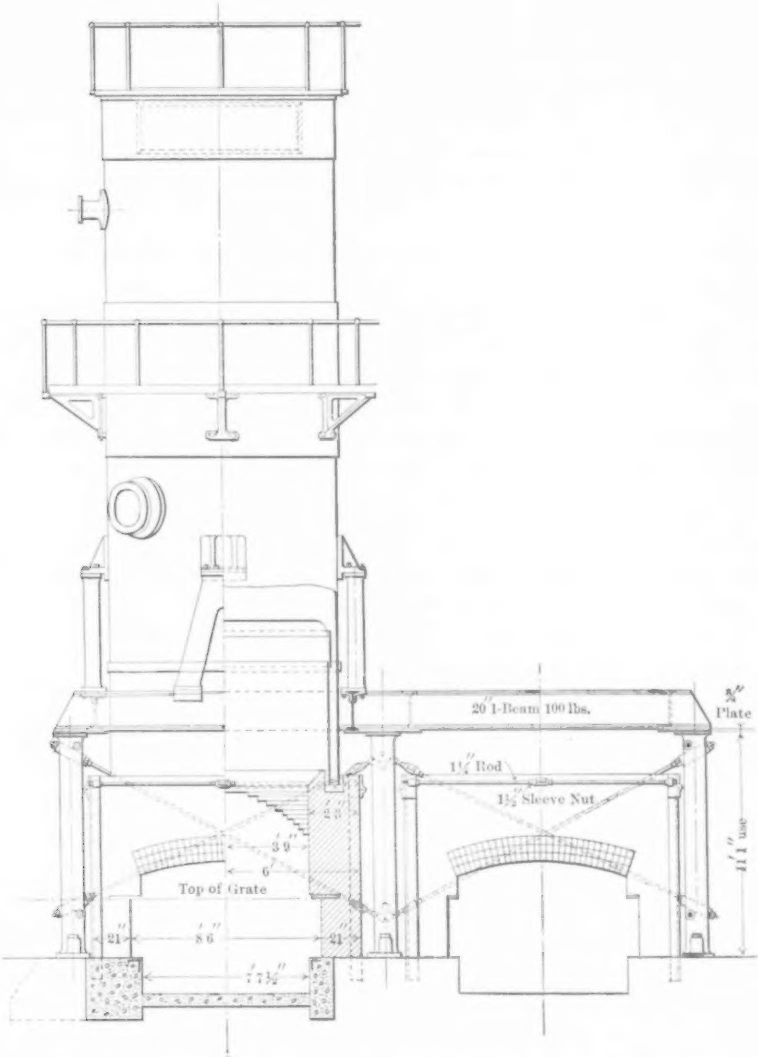


FIG. 2 SECTIONAL ELEVATION OF FURNACE OF THE AUTHOR'S FIRE-TUBE BOILER

discussion, it should be remembered that it was good, only not as good as is sometimes the case.

5 Mr. Parker states that the tubes leaked for the reason that they were set close to the grate and were therefore subjected to wide ranges

of temperature. This is true if we consider the closing of many of the tubes by clinker and the consequent overheating of those that were not closed.

6 I agree with Mr. Bement that some other kinds of stoker would probably not have precipitated the clinker on the tube ends, and this I stated in the paper.

7 Concerning the ability of the water to enter among the tubes, there are many large vertical boilers, some nearly as large as the one described, that have far less space for the passage of water among the tubes, and no trouble results. I know of some that have only one wide space across the crown sheet, while mine have eight wide spaces entirely across, or sixteen reaching to the center.

8 I observe that Mr. Bement considers that the cause of the cessation of the leakage of the tubes of my boiler was the added surface of the water leg. I cannot feel that this is so. It is inconceivable to me that the heat near the center of the furnace should be sensibly reduced thereby. Moreover the absence of the clinker after the change seems to me ample cause of the improvement, for, as I have stated in the paper, a large proportion of the tubes were stopped up, and those that were in service must have been overheated. I think that if the boilers had been raised without adding to the water leg the trouble would have ceased.

9 Whatever the cause of the leakage may have been, I find on

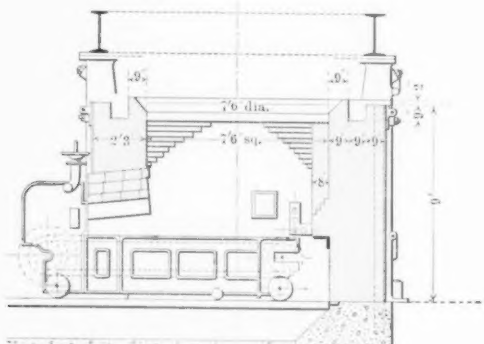


Fig 3 Section of Furnace of the Boiler Shown on page 279

January 17, 1910, the date of writing, that the tubes are not leaking, nor have they leaked since August 31, 1908, in the case of one boiler, and February 25, 1909, in the other, each boiler having been worked constantly to about 1000 boiler horsepower.

ACCESSIONS TO THE LIBRARY

This list includes only accessions to the library of this Society, included in the Engineering Library. Lists of accessions to the libraries of the A.I.E.E. and A.I.M. E. can be secured on request from Calvin W. Rice, Secretary, Am.Soc.M.E.

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Elements of Machine Design. By D. S. Kimball and J. H. Barr. *New York, J. Wiley & Sons, 1909.*

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Public Water Supplies. Edition 2. By F. E. Turneaure and H. L.Russell. *New York, J. Wiley & Sons, 1909.*

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Views and Description of Filtration System of the Denver Union Water Co.

Views and Description of Lake Cheesman. The Bulwark of Denver's Water Supply.

Welding and Cutting Metals by Aid of Gases or Electricity. By L. A. Groth. *New Yokr, Van Nostrand Co., 1909.*

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Carnegie Scholarship Memoirs. Vol. 1. *London, 1909.*

Fuel Tests With House-Heating Boilers. University of Illinois, Engineering Experiment Station. Bulletin No. 31. By J. M. Snodgrass. *Urbana, Ill., 1909.*

Incorporated Institution of Automobile Engineers. *Proceedings.* Vol. 3. *London, 1909.*

U. S. Steam Engineering, Bureau of. Annual Report. 1909. *Washington, 1909.*

TRADE CATALOGUES

American Radiator Company, *Chicago, Ill.* Stock list of boilers, radiators, etc., 283 pp.; The Houses Successful, 40 pp.; Ideal Heating, 46 pp.; Results Successful, 20 pp.; Ideal boilers, 32 pp.; Vento cast-iron, hot-blast heaters, 48 pp.; Ideal round boilers, 32 pp.; Ideal sectional boilers, 32 pp.

Newman Clock Comapny, *Chicago, Ill.* Testimonials concerning the watchmen's portable clok, 9 pp.

Niles-Bement-Pond Company, *New York, N. Y.* Pond rigid turret lathe, 44 pp.

Sanders, Rehders & Company, Ltd., *London, England.* Sarco patent gas volume recorder, 4 pp.; Recording and indicating steam meters, 4 pp.; Fuel calorimeters, 4 pp.; Patent reversion recorder, 2 pp.; Recording draught gauge, 4 pp.

Underfeed Stoker Company of America, *Chicago, Ill.* Publicity Magazine, December 1909, concerning the Jones Stoker, 16 pp.

Wisconsin Engine Company, *Corliss, Wis.* Bulletin C-4. Heavy Duty Corliss Engines, belted type, 24 pp.

UNITED ENGINEERING SOCIETY

Index to Economic Material in Documents of Maine, 1820-1904; Massachusetts, New Hampshire, New York, Rhode Island, Vermont, 1789-1904. By A. R. Hasse. *Washington, 1907-1908.*

Manufacturers' Record. Annual Blue-book of Southern Progress, 1909. *Baltimore, 1909.* Gift of Manufacturers' Record.

Illustrations of Concrete Steel Arch Bridges. Gift of Concrete Steel Engineering Company.

EMPLOYMENT BULLETIN

The Society has always considered it a special obligation and pleasant duty to be the medium of securing better positions for its members. The Secretary gives this his personal attention and is most anxious to receive requests both for positions and for men available. Notices are not repeated except upon special request. Copy for notices in this Bulletin should be received before the 15th of the month. The list of men available is made up of members of the Society and these are on file, with the names of other good men not members of the Society, who are capable of filling responsible positions. Information will be sent upon application.

POSITIONS AVAILABLE

08 A company in the Middle West manufacturing moderately heavy machinery has a vacancy for an expert mechanical designer. The position will require a man of considerable experience in the designing of power machinery, familiar with steam and compressed air, and the application of electric motors. Must be capable of submitting original designs to meet conditions. Give experience fully, salary expected, and quote references. Location, Middle West.

09 Mechanical engineer, familiar with the designing and building of steam shovels. Location, Toronto, Canada.

MEN AVAILABLE

14 Technical graduate, experienced in several varied lines of industry, holding executive positions of responsibility during the last eight or nine years, desires to become associated in position of trust with good manufacturing concern, preferably located in the East or Middle West. Best of references.

15 Member, graduate Stevens Institute, eighteen years experience in design and construction of power plants and special machinery; competent to prepare plans, specifications and estimates; desires position with first-class firm of consulting engineers or manufacturing concern.

16 Graduate electrical and mechanical engineer, thirteen years practical experience in testing, inspection and construction work; past five years in charge electrical department and power plant of industrial establishment; executive ability and excellent references; desires to change to a broader field. Qualified for assistant to consulting engineer, assistant superintendent of manufacturing plant or assistant manager. Salary $2500.

17 Stevens graduate '97, extensive shop and drawing room experience, including steel, reinforced concrete, power house and conveying installations. New York city preferred.

18 Associate, experienced in design and installation of mechanical equipment of power stations, railway and industrial buildings, desires position in Middle West; competent executive in field or drafting room. Salary $200 per month, or on percentage basis with consulting engineers.

19 General manager, or assistant, graduate M. E., at present holding similar position, would like to make change; ten years practical experience; good executive ability, best of references.

20 Member, thoroughly experienced in the design of large gas engines for all services; desires position as chief engineer with company building this class of machinery or with a company desiring to enter the field, preferably the latter.

21 Junior, age twenty-nine, technical graduate, seven years experience shop, foundry, drawing room and office: desires to make a change. Compressed air, pumping machinery, or similar line preferred.

22 Junior member, technical graduate, nine years experience in drafting, construction and office work with engineers and contractors, in engineering departments of industrial companies; wishes to make change after May first. Prefer position with firm of engineers and contractors or with industrial company.

23 Student member, graduating Cornell University June 1910, desires position with engineering concern in New York city or thereabouts. Salary no object. Can furnish highest references.

24 Graduate of M.I.T., in electrical and mechanical engineering; experience in design and construction of machinery and buildings; development of systems and organization; making of reports, compilation of data, etc.

COMING MEETINGS

FEBRUARY-MARCH

Advance notices of annual and semi-annual meetings of engineering societies are regularly published under this heading and secretaries or members of societies whose meetings are of interest to engineers are invited to send such notices for publication. They should be in the Editor's hands by the 18th of the month preceding the meeting. When the titles of papers read at monthly meetings are furnished they will also be published.

AMERICAN ASSOCIATION OF RAILROAD SUPERINTENDENTS
March 18, Chicago. Secy., O. G. Fetter.

AMERICAN INSTITUTE OF ELECTRICAL ENGINEERS
February 11, monthly meeting, 29 W. 39th St., New York. Paper: A Modern Automatic Telephone Apparatus, W. Lee Campbell. Secy., R. W. Pope.

AMERICAN INSTITUTE OF MINING ENGINEERS
March 1-5, Spring meeting, Hotel Shanley, Pittsburg, Pa. Secy., R. W. Raymond, 29 W. 39th St., New York.

AMERICAN MATHEMATICAL SOCIETY
February 26, New York and San Francisco sections. Secy., F. N. Cole, 501 W. 116th St., New York.

AMERICAN RAILWAY ENGINEERING ASSOCIATION
March 14-17, Chicago. Secy., E. H. Field, Monadnock Bldg.

AMERICAN SOCIETY OF CIVIL ENGINEERS
February 2, 16, 220 W. 57th St., New York, 8:30 p.m. Papers: Underpinning the Cambridge Building, New York City, by T. K. Thomson; Effect of Alkali on Concrete, by G. G. Anderson. Secy., C. W. Hunt.

AMERICAN SOCIETY OF ENGINEERING CONTRACTORS
February 24-26, annual meeting, Chicago, Ill. Secy., Daniel J. Haner, Park Row Bldg., New York.

AMERICAN SOCIETY OF MECHANICAL ENGINEERS
February 8, 29 W. 39th St., New York. February 16, City Club, Boston, May 31-June 3, Spring Meeting, Altantic City, N. J. July 26-29, joint meeting with Institution of Mechanical Engineers, Birmingham, England. Secy., Calvin W. Rice, 29 W. 39th St.

ASSOCIATION OF ONTARIO LAND SURVEYORS
February 22-24, annual meeting. Secy., Killaly Gamble, 703 Temple Bldg., Toronto.

BOSTON SOCIETY OF ARCHITECTS
February 1, Parker House, Boston, Mass. Dinner in the Crystal Room at 6.30 p. m. Secy., Edwin J. Lewis, Jr.

CANADIAN FORESTRY ASSOCIATION
March 10-11, Fredericton, N. B. Secy., Jas. Lawler, 11 Queen's Park, Toronto, Ont.

CANADIAN MINING INSTITUTE

March 2-4, annual meeting, Toronto, Ont. Secy., H. Mortimer-Lamb, Windsor Hotel, Montreal.

CONNECTICUT SOCIETY OF CIVIL ENGINEERS

February 8, annual meeting, New Haven, Conn. Secy., J. Frederick Jackson, Box 1304, New Haven, Conn.

ENGINEERING SOCIETY OF WISCONSIN

February 23-25, Milwaukee, Wis. Secy., W. G. Kirchoffer, 31 Vroman Bldg., Madison.

ENGINEERS CLUB OF PHILADELPHIA

February 5, annual meeting, 1317 Spruce St. Secy., W. P. Taylor.

INSTITUTION OF MECHANICAL ENGINEERS

February 18, Institution House, Storey's Gate, St. James' Park, Westminster, S. W., London, England. Secy., Edgar Worthington.

IOWA ASSOCIATION CEMENT USERS

March 9-11, Cedar Rapids. Secy., Ira Williams, Ames.

IOWA ENGINEERING SOCIETY

February 16-17, Cedar Rapids. Secy., A. H. Ford, Iowa City.

MINNESOTA ELECTRIC ASSOCIATION

March, St. Paul. Secy., B. W. Cowperthwait.

NATIONAL ASSOCIATION OF CEMENT USERS

February 21-26, annual convention, Chicago, Ill. Pres., Richard L. Humphrey, Harrison Bldg., Philadelphia, Pa.

NEBRASKA CEMENT USERS ASSOCIATION

February 1-4, Lincoln. Secy., Peter Palmer, Oakland.

NEW ENGLAND ASSOCIATION OF GAS ENGINEERS

February 16, 17, annual meeting, Boston, Mass. Secy., N. W. Gifford, 26 Central Sq., East Boston, Mass.

NEW ENGLAND RAILROAD CLUB

March 8, annual meeting, Boston, Mass. Secy., George H. Frazier, 10 Oliver St.

NEW ENGLAND STREET RAILWAY CLUB

March 24, annual meeting, Boston, Mass. Secy., J. J. Lane, 12 Pearl St.

NEW ENGLAND WATERWORKS ASSOCIATION

February 9, Hotel Brunswick, Copley Sq., Boston. Papers: Depreciation, L. G. Powers; The Purchase of Coal on Efficiency Basis, A. O. Doane.

NORTHWESTERN CEMENT PRODUCTS ASSOCIATION

February 18-21, annual meeting, Chicago, Ill. Chairman notification committee, O. U. Miracle, Minneapolis, Minn.

PACIFIC COAST ELECTRIC AUTOMOBILE ASSOCIATION

February, Oakland, Cal. Secy., J. T. Halloran, 604 Mission St., San Francisco.

PROVIDENCE ASSOCIATION OF MECHANICAL ENGINEERS

February 15, West Hall of the Rhode Island School of Design, 8 p. m. Paper: The Bliss-Levitt Torpedo, Samuel Aronson and Chas. Gabriel.

RAILWAY SIGNAL ASSOCIATION

March 14, Chicago. Secy., C. C. Rosenberg, Bethlehem, Pa.

SCRANTON ENGINEERS CLUB

February 3, annual dinner, Club Rooms. Secy., A. B. Dunning.

SOUTHERN GAS ASSOCIATION

February 16, Chattanooga, Tenn. Secy., James Ferrier, Rome, Ga.

STEVENS ENGINEERING SOCIETY

February 8, 15, 21, Hoboken, N. J. Papers: Pavements for City Streets, Samuel Whinery, Mem. Am. Soc. M. E.; Power Plant Economies, D. S. Jacobus, Mem. Am. Soc. M. E.; The Engineer as a Manager, H. L. Gantt, Mem. Am. Soc. M. E. Secy., R. H. Upson.

UNIVERSITY OF CINCINNATI, Student Branch, AM. SOC. M. E.

February 18, regular meeting. Paper: Milling Machines and their Uses, C. S. Gingrich, Jun. Am. Soc. M. E. Secy., P. G. Haines.

MEETING IN THE ENGINEERING SOCIETIES BUILDING

Meetings of All Kinds

Date	Society	Secretary	Time
February			
2	Wireless Institute	S. L. Williams	7.30
3	Blue Room Engineering Society	W. D. Sprague	8.00
5	Amer. Soc. Hungarian Engrs. and Archs	Z. de Németh	8.30
8	The American Society of Mechanical Engrs	Calvin W. Rice	8.15
10	Illuminating Engineering Society	P. S. Millar	8.00
11	American Institute Electrical Engineers	R. W. Pope	8.00
15	New York Telephone Society	T. H. Lawrence	8.00
18	New York Railroad Club	H. D. Vought	8.15
23	Municipal Engineers of New York	C. D. Pollock	8.15
March			
2	Wireless Institute	S. L. Williams	7.30
3	Blue Room Engineering Society	W. D. Sprague	8.00
5	Amer. Soc. Hungarian Engrs. and Archs	Z. de Németh	8.30
8	The American Society of Mechanical Engrs	Calvin W. Rice	8.15
10	Illuminating Engineering Society	P. S. Millar	8.00
11	American Institute Electrical Engineers	R. W. Pope	8.00
15	New York Telephone Society	T. H. Lawrence	8.00
18	New York Railroad Club	H. D. Vought	8.15
23	Municipal Engineers of New York	C. D. Pollock	8.15

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1909

On a Standard Tonnage Basis for Refrigeration

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On Constitution and By-Laws

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On Conservation of Natural Resources

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On Meetings of the Society in St. Louis

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F. W. TAYLOR

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Stevens Inst. of Tech., Hoboken, N. J.	December 4	Alex. C. Humphreys	H. H. Haynes	R. H. Upson
Cornell University, Ithaca, N. Y.	December 4	R. C. Carpenter	....	C. F. Hirshfeld
	1909			
Armour Inst. of Tech., Chicago, Ill.	March 9	C. F. Gebhardt	N. J. Boughton	M. C. Shedd
Leland Stanford, Jr. University, Palo Alto, Cal.	March 9	W. F. Durand	E. A. Rogers	H. C. Warren
Polytechnic Institute, Brooklyn, N. Y.	March 9	W. D. Ennis	J. S. Kerins	Percy Gianelia
State Agri. College, Corvallis, Ore.	March 9	Thos. M. Gardner	C. L. Knopf	S. H. Graf
Purdue University, Lafayette, Ind.	March 9	L. V. Ludy	E. A. Kirk	J. R. Jackson
Univ. of Kansas, Lawrence, Kan.	March 9	P. F. Walker	H. S. Coleman	John Garver
New York Univ., New York	November 9	C. E. Houghton	Harry Anderson	Andrew Hamilton
Univ. of Illinois, Urbana, Ill.	November 9	W. F. M. Goss	W. F. Colman	S. G. Wood
Penna. State College, State College, Pa.	November 9	J. P. Jackson	G. B. Wharen	G. W. Jacobs
Columbia University, New York	November 9	....	F. R. Davis	H. B. Jenkins
Mass. Inst. of Tech., Boston, Mass.	November 9	Gaetano Lanza	Fredk. A. Dewey	A. P. Truette
Univ. of Cincinnati, Cincinnati, O.	November 9	J. T. Faig	W. H. Mantgomery	P. G. Haines
Univ. of Wisconsin, Madison, Wis.	November 9	C. C. Thomas	R. N. Trane	G. A. Glick
Univ. of Missouri, Columbia, Mo.	December 7	H. Wade Hibbard	R. E. Dudley	F. T. Kennedy
Univ. of Nebraska, Lincoln, Neb.	December 7	C. R. Richards	....	....